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SCHEDULING THE WORK OF THE DRAFTING ROOM.

THE APPLICATION OF SCHEDULING IN THE DESIGN SECTION OF THE BROOKLYN NAVY YARD.

BY ABRAHAM SOLOV.*

INTRODUCTION.

The building of a Naval vessel is a complex process requiring complete coordination between the various departments in the Yard. Plans, material and labor must be marshalled into an orderly, well integrated and continuously flowing process having for its end purpose the efficient construction of a complicated, well balanced machine—the modern warship.

Scheduling is the tool by means of which this essential coordination is achieved. Each activity involved in the construction of the vessel must be scheduled individually and then integrated into a master schedule showing the relation between it and the rest of the Yard.

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Before going into the subject of Schedules it might be well to outline briefly the portion of the Yard organization which concerns us here.

The construction and repair of vessels is under the cognizance of the Industrial Department of the Navy Yard. The Production Division of this Department concerns itself with the actual construction and shopwork. The procurement of material is handled by the Material Section of the Planning Division. The Planning Division, through the Design Section, also prepares the plans and performs the engineering required in the construction of the vessel.

The Design Section consists of several subdivisions, namely: Ordnance Design, Electrical Design, Hull Design, and Machinery Design. This paper deals with the scheduling procedure carried on in the Machinery Design subdivision which has jurisdiction over the machinery, piping and auxiliaries forming the ship's power plant.

Machinery Erection, Material Procurement and Design Schedule.

In scheduling the design and construction of a vessel building in a government Yard the following major events have to be considered, in inverse order of occurrence, from the time the building period begins:

1. Date erection will begin on board ship.
2. Time required to fabricate the raw material.
3. Material procurement.
   a. Time required by the manufacturer or vendor to supply the equipment or material.
   b. Time required by the Bureau of Supplies and accounts to approve the requisition, advertise for bids and award the contract.
   c. Time required to prepare requisitions.
4. Design Period.

All of these factors are shown in relation to each other on a chart which we call the "Machinery Erection, Material Procurement and Design Schedule". This schedule is illustrated in Figure 1. You will note that this is a modified form of Gantt Chart.

The preparation of this schedule is begun as soon as the instruc-

tions for the design and construction of a vessel are received. The specifications and contract plans are carefully examined and analyzed to determine what piping systems and equipment, under cognizance of the Machinery Division, will be required for the vessel. These items are then listed and sent to the Production Division.

In the Production Division the list is first turned over to the Ship Superintendent who assigns the dates on which erection is to begin for each of the items listed. Then the Shop Superintendent assigns to each item the time the Yard will require to fabricate the material. The list with the above information is then sent back to the Machinery Drafting room where the schedules are prepared.

It should be noted that in preparing this schedule we begin by setting our goal first, i.e. the start of erection. This is the only way that we can make a schedule which will serve our purpose. When a ship is contracted for, that is, assigned for construction at the Yard, the Secretary of the Navy sets the date when the vessel is to be completed. All our departments therefore must schedule their work so that the erection of the vessel and its machinery is accomplished in such a way that this date is met.

The Ship Superintendent knowing his facilities and methods of construction is in the best position to determine the sequence, and date of start of erection required for each component, in order that the construction of the vessel may proceed in an orderly and efficient manner, and at the same time, meet the completion date set. That is why the list of items is sent to him first.

Having the required erection dates and knowing the fabrication time required, we then know exactly when plans and material must be available. After the list of items is received from the Production Division the Material Procurement Section is consulted as to their requirements. From previous experience modified of course by contemporary conditions, the interval for the delivery of the raw material, or the manufacture and delivery of the item by the vendor is estimated, and the date on which the contract should be let is thus established.

Working back from this point we lay off the time required by the Bureau to approve the requisition, advertise the bids and award the contract as indicated by the dotted lines, Figure 1.

This brings us to the point where the requisition must be completed. The period required by the Material Section to prepare the requisition is estimated and then indicated on the Chart Figure 1.

Since most plans are required before requisitions can be written, the start of the requisition date is the deadline for plan completion. Detail and other plans not required for ordering materials may be completed not later than the date when shop fabrication is to start. The Navy Yard practice however is such that the majority of plans drawn are used for ordering material, and for that reason we indicate the end of the design period at the start of requisition preparation.

It is obvious that we cannot start the design of any piping system any time we please, since the information needed may not be available before certain contracts are let, or before the design of some other system or piece of equipment has advanced to a certain point. The design starting points then have to be estimated on the basis of past experience and on information available at the time the schedule is being prepared. We arrive thus at a period defined by two limits during which we must design any particular piping system or equipment. These design periods are indicated for each major item by the single solid line in Figure 1.

This main schedule, once it is agreed upon, is never changed except due to some major cause such as a change in the required completion date or some other equally important reason.

During the present emergency the procedure indicated above has to be modified in order to conform to the requirements imposed by the Bureau of Ship's Component Percentage and Piping Schedules. The effect of these Bureau schedules is to shift our governing date (or starting point) from Erection to that date on which material would be received in the Yard. Instead of starting from the Erection date and working to the left (Figure 1), we now have to start at the date obtained from the Bureau schedules and work to the left and right of it.

An article explaining the "Bureau Schedules" and their effect on the building program appeared in the November 1944 issue of this JOURNAL.*

* JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. 56, No. 4, November 1944, "Ship Material Scheduling" by Herbert M. Neuhaus.

Plan Schedules.

As soon as the Main Schedule is approved and agreed to by all concerned, and the design periods are set, then we set about scheduling the work of the Machinery Division in detail. For this purpose we use the chart illustrated in Figure 2. This chart is also a form of Gantt chart adopted for our special purpose. The Machinery Division is divided into several sections. Each section is responsible for the design of several systems or groups of equipment. One or more of the charts, Figure 2, as required, are used for the scheduling of each section's work.

In the preparation of these plan schedules, all the plans that will be needed for the design of the ship are listed in advance. This list is complete, and indicates the exact titles of the plans. This requires a little thought but it is not too difficult if you know your job; particularly when a similar type vessel has been designed before. Of course this list cannot be 100 per cent correct and as the job progresses additions and subtractions may have to be made from time to time. However, the changes are comparatively few and the list is complete enough for scheduling purposes.

The listed plans are classified and grouped under the major divisions listed in Figure 1. Plan numbers are assigned and the plans are entered on the Plan Schedule, Figure 2. The column on the extreme left of Figure 2 indicates the item number under which the plans in any group are scheduled. This item number corresponds to the item number assigned to the group on the Main Erection Schedule, Figure 1. In this way it is possible to correlate the Plan and Main Schedules.

The periods of design, material procurement and shop fabrication, as well as the start of erection, as indicated on the Main Schedule are also indicated on the Plan Schedules for easy reference.

We are now ready to assign dates to each plan individually. This is done in such manner as to insure that each plan is worked on and completed in a sequence that will result in a reasonably constant manpower load, and at the same time assure completion of all the plans by the required time. Therefore, each plan is assigned not only a completion date but also a starting date, and the number of man-months (or man-days) work required to complete it.

Each shaded area in Figure 2 represents the design period for the corresponding plan. Each heavy horizontal line superimposed on the shaded area represents one man working during that period. The length of this horizontal line represents the estimated time in man-months (or man-days) required to be put into the plan. The line need not extend from beginning to end of the shaded area but could cover only a portion of it. Such a condition would occur where we have a plan that would require intermittent work on it for some reason or other.

Take, for example, plan number S5501-21 in Figure 2. It is estimated that if one man worked on it continuously it would take him three weeks to complete it. In other words it would require 18 man-days (6 man-days per week) to turn out the design. However, only part of the information needed to complete the plan is available at the start and during the early stages of the design period, and, on the other hand, some other design is dependent on the information this plan will give so that some work has to be done on it to keep things moving. In this case, then, the man will have to work on the plan intermittently. Graphically this is shown by the shaded area extending from March 15, 1943 to April 15, 1943 and the horizontal line extending for only three weeks during that period.

The progress of each plan is shown in the column marked "per cent completion." This column is divided into ten equal spaces and the percentage completion is indicated by the terminus of the horizontal line opposite each plan. Progress is recorded at stated intervals, usually monthly on new construction or long term projects, and at shorter intervals for short term projects.

The status of the plans is obtained from progress reports which each Section Chief is required to submit at the end of each month (or more often if required). This report lists all plans scheduled for completion within the six months following the date of the report, and any additional plans that are being worked on.

Plans reported completed and issued are double checked by going through the invoices of the Dispatching Section, which distributes the plans to the various activities. A plan is not considered one hundred per cent complete until it is actually issued from the Design Section. The date on which a plan is issued is indicated on the plan schedule by a check mark, see Figure 2.

ITEM NO.	TITLE
1	ARRANGEMENT OF MACHINERY
2	ARRANGEMENT OF FOUNDATION BOLTING
3	HEATING COILS IN F.O. TANKS FORD. OF FRAME #36
4	STEAM DRAIN COLLECTING UNIT
5	DISTILLING UNITS (EXCLUSIVE OF PUMPS)
6	HEATING COILS IN F.O. TANKS ABREAST OF MACHY SPACES
7	BOILERS - UNITS FURNISHED COMPLETE
8	FEED WATER HEATERS - DEAERATING & STORAGE SECTIONS
9	HIGH PRESSURE AIR COMPRESSORS
10	MEDIUM PRESSURE AIR COMPRESSORS
11	MAIN INJECTION & OVERBOARD DISCHARGE VALVES
12	HEATING COILS IN F.O. TANKS AFT OF MACHY SPACES
13	PUMPS FOR DISTILLING PLANT
14	F.O. PIPING IN MACHY SPACES
15	500 K.W. GENERATOR - DIESEL
16	F.O. PIPING FORWARD OF MACHY SPACES
17	F.O. PIPING AFT OF MACHY SPACES
18	REFRIGERATING MACHINERY
19	BEARINGS & STUFFING BOXES - STERN TUBE, PROP. & LINE SHAFTS
20	DISTILLING PLANT PIPING
21	BILGE PUMPS
22	MAGAZINE COOLING EQUIPMENT & PIPING 5" & 4" MAGAZINES
23	DYNAMO CONDENSER
24	FORCED DRAFT BLOWERS AND SHUTTERS
25	TURBO-GENERATORS
26	EMERGENCY FEED PUMPS
27	AIR COOLING EQUIPMENT & PIPING (16" MAG. FORD.)
28	LUB. OIL TANKS
29	MAIN FEED PUMPS
30	DYNAMO CONDENSER CONDENSATE PUMPS
31	MAIN CIRCULATING PUMPS
32	AUXILIARY FEED BOOSTER PUMPS
33	FEED PIPING
34	SALT WATER PIPING
35	MAIN FEED BOOSTER PUMP
36	MAIN CONDENSATE PUMP
37	F.O. BOOSTER & TRANSFER PUMPS
38	FUEL OIL SERVICE PUMPS
39	FIRE AND FLUSHING PUMPS
40	MAIN TURBINES AND CONDENSERS
41	MAIN REDUCTION GEAR
42	CONDENSATE & AIR EJECTOR PIPING
43	AIR ACCUMULATOR TANKS
44	BOTTOM BLOW PIPING
45	MAIN [illegible] PIP

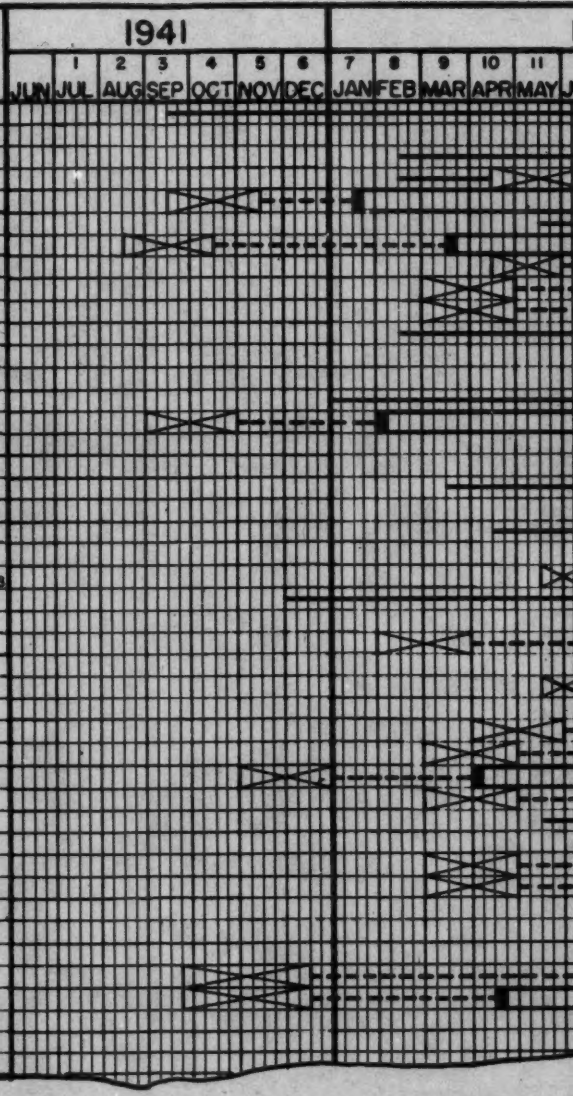
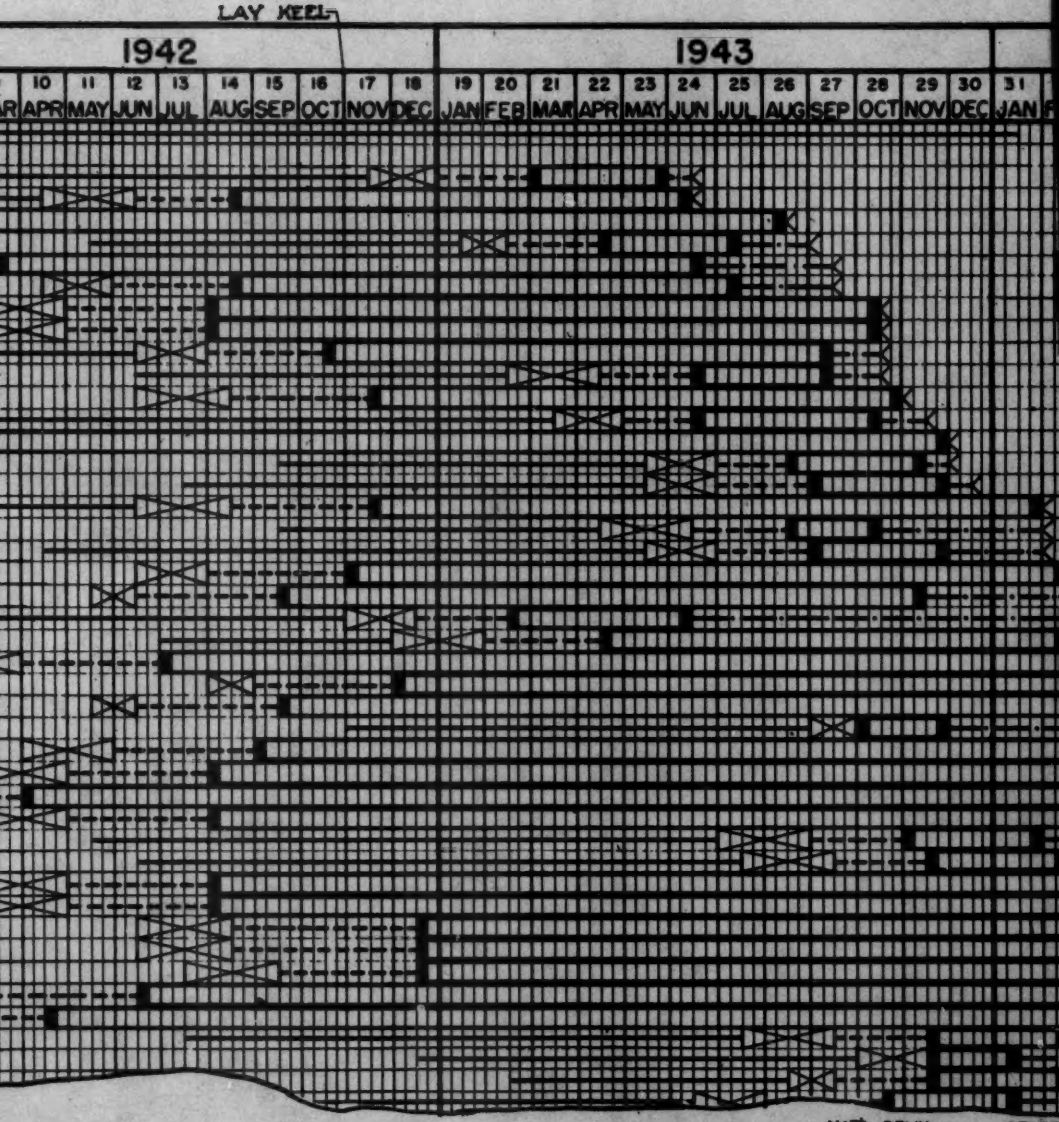


FIG. 1

MACHINERY ERECTION, MATERIAL
AND DESIGN SCHEDU



MATERIAL PROCUREMENT
N SCHEDULE

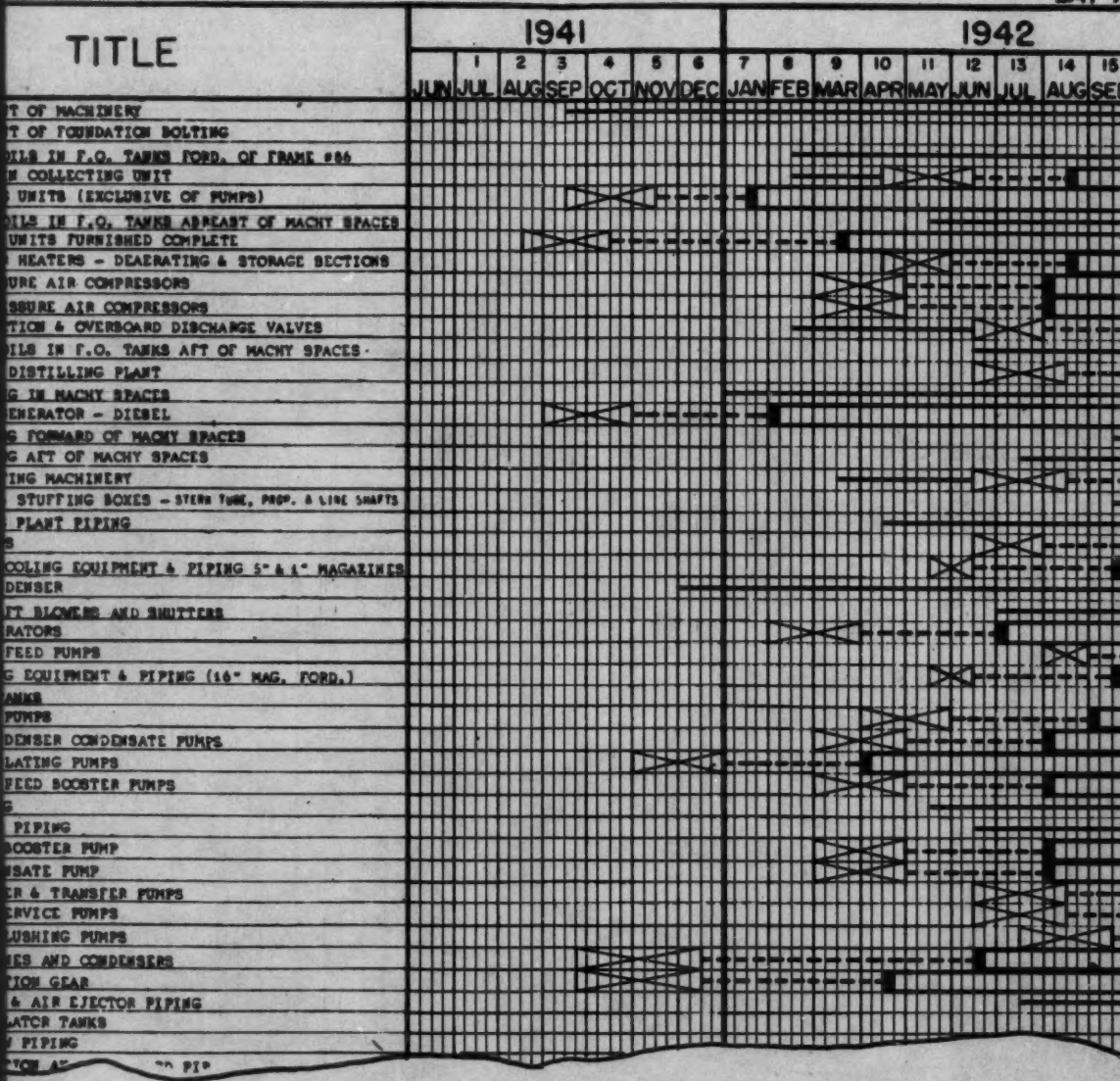
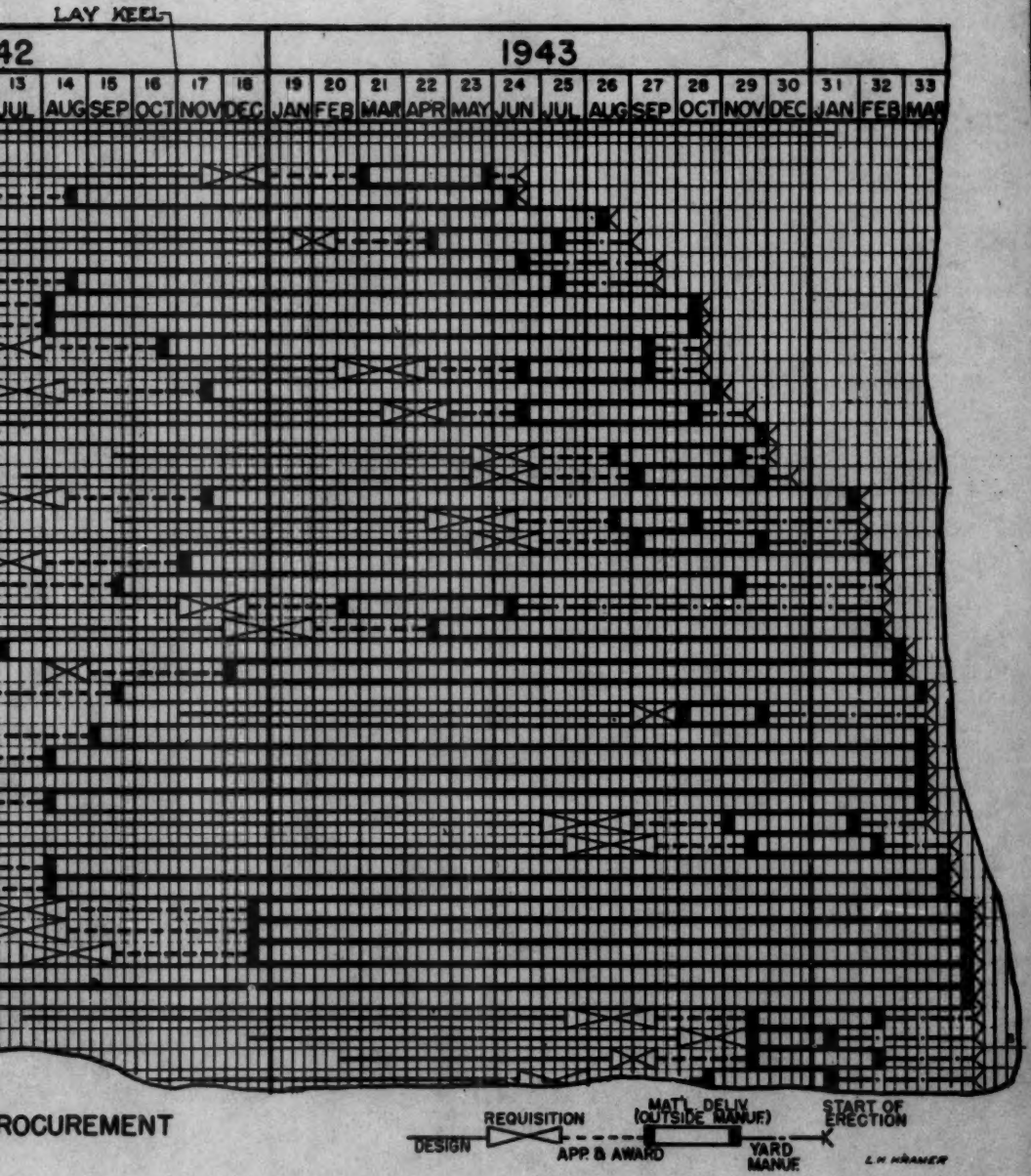


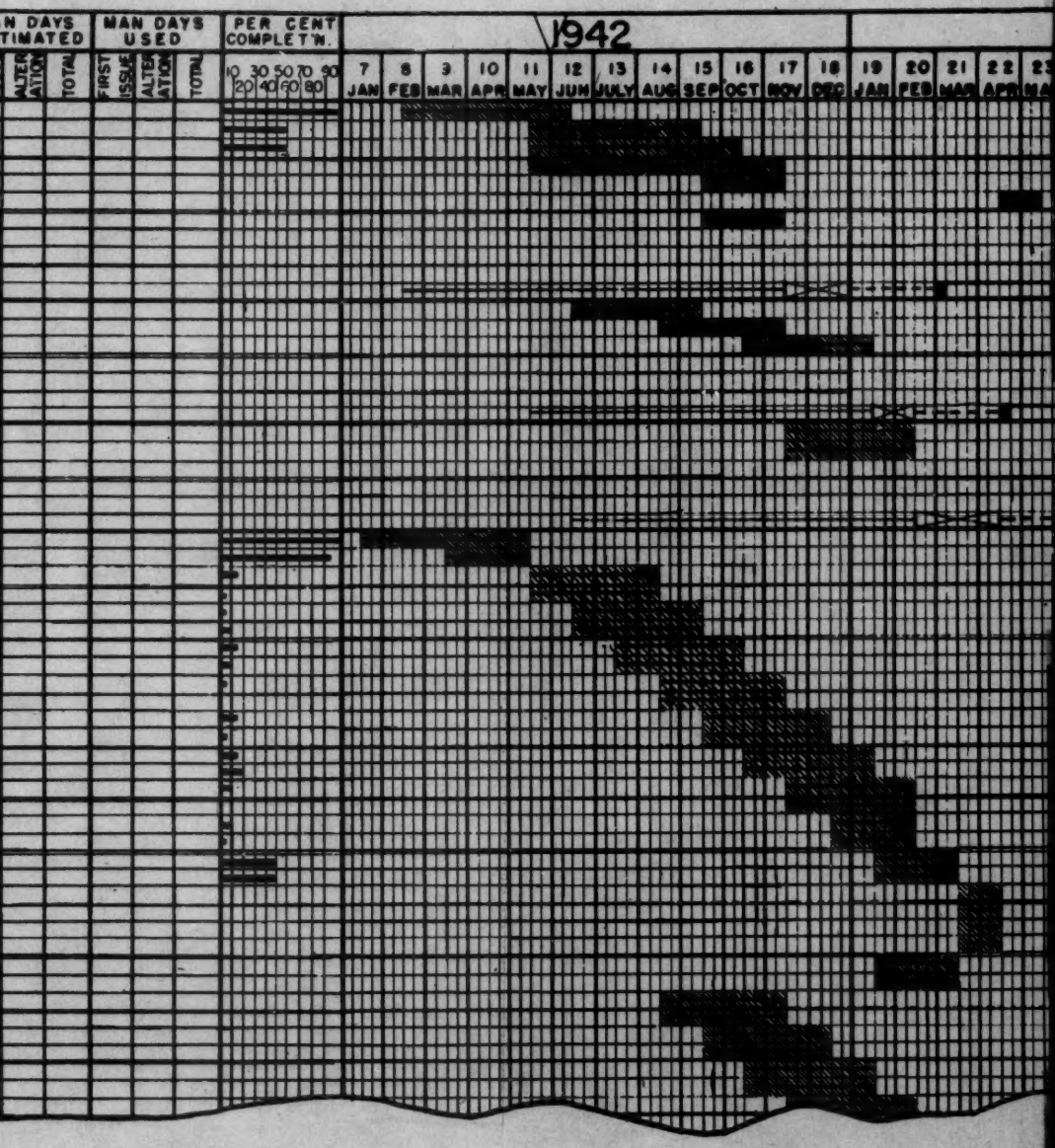
FIG. 1

MACHINERY ERECTION, MATERIAL PROCUREM
AND DESIGN SCHEDULE


LAY KEEL
42
1943
13 JUL
14 AUG
15 SEP
16 OCT
17 NOV
18 DEC
19 JAN
20 FEB
21 MAR
22 APR
23 MAY
24 JUN
25 JUL
26 AUG
27 SEP
28 OCT
29 NOV
30 DEC
31 JAN
32 FEB
33 MAR
ROCUREMENT
DESIGN
REQUISITION
APP. & AWARD
MAT'L DELIV (OUTSIDE MANUF.)
YARD MANUF.
START OF ERECTION
L.H. HRAMER

ITEM NO.	TITLE OF PLAN	PLAN NO	MAN DAYS ESTIMATED FIRST ISSUE	ALTERATION	TOTAL	MA… FIRST…
	F.O. SYSTEM-HEATING COILS-DIAGR. ARRGT.	S5503-2				
	" " " " " -ARRANGEMENT-PLAN & B/M - FR. #39-58	S5503-3				
	" " " " " " " -SECT. & ELEV. FR. #39-58	S5503-4				
	" " " " " " " -FR. #58-72	S5503-5				
	" " " " " -STEAM & DRAIN MANIFOLDS-DETS.	S5503-9				
3	" " " " " - " " " " -HANDWHEEL MARKINGS	S2803-8				
	" " " " " -DETAILS OF FITTINGS	S5503-10				
	F.O. SYSTEM-HEATING COILS-ARRGT.-FR. #72-100	S5503-6				
	" " " - " " - " -FR. #100-120	S5503-7				
6	" " " - " " - " -FR. #120-140	S5503-8				
	F.O. SYSTEM-HEATING COILS-ARRGT.- PLAN & B/M AFT. FR. #140	S5503-11				
	" " " - " " - " -SECTS. & ELEV. AFT. FR. #140	S5503-12				
12						
	F.O. SYSTEM-FILLING, TRANSF. SERV. SUCT. & DISCH. PIPING-DIAG. ARRGT.	S5501-1				
	" " " -TANK DRAIN PIPING-DIAG. ARRGT.	S5501-2				
	" " " -FILLING & TRANSFER & TANK DRAIN PIPING-ARRGT.-ENG. RM. #1	S5501-3				
	" " " - " " " " " " " " - " - " " #4	S5501-4				
	" " " - " " " " " " " " - "AFT. DIESEL GEN.&AIR COMPR. RM.	S5501-5				
	" " " - " " " " " " " " - "AFT. EVAP. & DEGAUSSING RM.	S5501-6				
	" " " - " " " " " " " " - "FWD. EVAP. & DIESEL GEN. RM.	S5501-7				
	" " " - " " " " " " " " - "FWD. AIR COMPR. RM.	S5501-8				
	" " " - " TRANSF. & SERV. SUCT. & TANK DRAIN PIPING-ARRGT.-FIRE RM. #2A	S5501-9				
	" " " - " " " " " " " " " " " " " #2B	S5501-10				
	" " " - " " " " " " " " " " " " " #3A	S5501-11				
	" " " - " " " " " " " " " " " " " #3B	S5501-12				
	" " " - " " " " " " " " " " " " " #1A	S5501-13				
	" " " - " " " " " " " " " " " " " #4A	S5501-14				
	" " " - " " " " " " " " " " " " " #1B	S5501-15				
	" " " - " " " " " " " " " " " " " #4B	S5501-16				
	" " " -FILLING & TRANSF. & TANK DRAIN PIPING-ARRGT.-DYNAMO & SWBD. RMS. #1	S5501-17				
	" " " - " " " " " " " - " - " " " " #2	S5501-18				
	" " " - " " " " " " " - " - " " " " #3	S5501-19				
	" " " - " " " " " " " - " - " " " " #4	S5501-20				
	(" " " - " " " " " " " -PIPE DETS.- SHEET #1	S5501-21)				
	" " " - " " " " " " " - " " - " #2	S5501-22				
14	" " " - " " " " " " " - " " - " #3	S5501-23				
	" " " - " " " " " " " - " " - " #4	S5501-24				
	" " " - " " " PIPING-MANIFOLDS - SHEET #1	S5501-25				
	" " " - " " " " - " - " #2	S5501-26				
	" " " -SERVICE PIPING - ARRGT. - FIRE ROOM #2A	S5501-27				
	" " " - " " " " " #2B	S5501-28				
	" " " - " " " " " #3A	S5501-29				
	" " " - " " " " " #3B	S5501-30				
	" " " - " " " " " #1A	S5501-31				
	" " " - " " " " " #4A	S5501-32				
	" " " " " #1B	S5501-33				

FIG. 2 PLAN



PLAN SCHEDULE

L.H. KRAMER

TITLE OF PLAN	PLAN NO	MAN DAYS ESTIMATED			MAN DAYS USED			PER COMP
		FIRST ISSUE	ALTERATION	TOTAL	FIRST ISSUE	ALTERATION	TOTAL	10 20 30 40
.O. SYSTEM-HEATING COILS-DIAGR. ARRGT.	S5501-2							
" " " " -ARRANGEMENT-PLAN & B/M - FR. #39-58	S5503-3							
" " " " " " -SECT. & ELEV. FR. #39-58	S5503-4							
" " " " " " -FR. #58-72	S5503-5							
" " " " -STEAM & DRAIN MANIFOLDS-DETS.	S5503-9							
" " " " " " " " " " -HANDWHEEL MARKINGS	S2803-8							
" " " " -DETAILS OF FITTINGS	S5503-10							
.O. SYSTEM-HEATING COILS-ARRGT.-FR. #72-100	S5503-6							
" " " " " " -FR. #100-120	S5503-7							
" " " " " " -FR. #120-140	S5503-8							
.O. SYSTEM-HEATING COILS-ARRGT.-PLAN & B/M AFT. FR. #140	S5503-11							
" " " " " " -SECTS. & ELEV. AFT. FR. #140	S5503-12							
.O. SYSTEM-FILLING, TRANSF. SERV. SUCT & DISCH. PIPING-DIAG. ARRGT.	S5501-1							
" " -TANK DRAIN PIPING-DIAG. ARRGT.	S5501-2							
" " -FILLING & TRANSFER & TANK DRAIN PIPING-ARRGT.-ENG. RM. #1	S5501-3							
" " " " " " " " " " " " " " " " #4	S5501-4							
" " " " " " " " " " " " " " "AFT. DIESEL GEN. & AIR COMPR. RM.	S5501-5							
" " " " " " " " " " " " " " "AFT. EVAP. & DEGAUSSING RM.	S5501-6							
" " " " " " " " " " " " " " "FWD. EVAP. & DIESEL GEN. RM.	S5501-7							
" " " " " " " " " " " " " " "FWD. AIR COMPR. RM.	S5501-8							
" " " " TRANSF. & SERV. SUCT. & TANK DRAIN PIPING-ARRGT.-FIRE RM. #2A	S5501-9							
" #2B	S5501-10							
" #3A	S5501-11							
" #3B	S5501-12							
" #1A	S5501-13							
" #4A	S5501-14							
" #1B	S5501-15							
" #4B	S5501-16							
" " -FILLING & TRANSF. & TANK DRAIN PIPING-ARRGT.-DYNAMO & SWBD. RMS. #1	S5501-17							
" #2	S5501-18							
" #3	S5501-19							
" #4	S5501-20							
" " " " " " " " " " " " " " -PIPE DETS.- SHEET #1	S5501-21							
" #2	S5501-22							
" #3	S5501-23							
" #4	S5501-24							
" " " " " " PIPING-MANIFOLDS - SHEET #1	S5501-25							
" #2	S5501-26							
" " -SERVICE PIPING - ARRGT. - FIRE ROOM #2A	S5501-27							
" " " " " " " " " " " " " " " " #2B	S5501-28							
" " " " " " " " " " " " " " " " #3A	S5501-29							
" " " " " " " " " " " " " " " " #3B	S5501-30							
" " " " " " " " " " " " " " " " #1A	S5501-31							
" " " " " " " " " " " " " " " " #4A	S5501-32							
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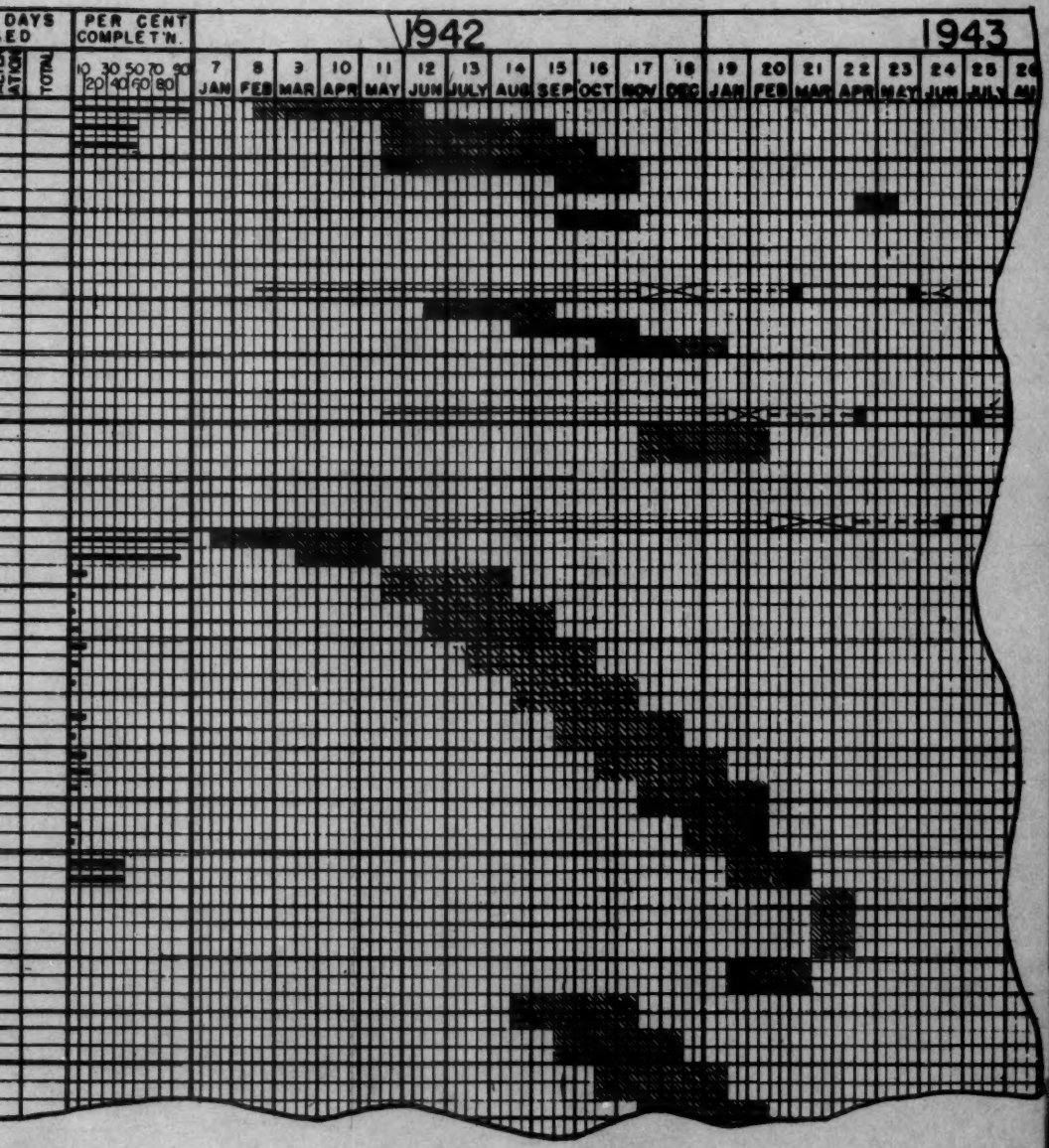
FIG. 2 PLAN SCHEDULE



SCHEDULE

L.H. KRAMER

In peace time many of the plans and designs prepared in the Yard have to be submitted to the Bureau of Ships in Washington for approval. At present, due to wartime conditions this requirement has been suspended and local approval is sufficient. When Bureau approval is required the symbol $\surd$ is inserted on the date on which the plan was sent to the Bureau and then, when the Bureau action has been received, the symbol $\rightarrow$ shows the date of receipt of such action.

As mentioned before the plan schedules are revised at stated intervals. The symbol $\vee$ placed just above the calendar scale indicates the latest revision date. Thus, the schedule illustrated in Figure 2 is correct to June 1, 1942. Revision of the plan schedules includes, besides the percentages and dates discussed above, also any additions, cancellations or changes in titles or plans.

In order to keep positive control of the schedules, a supervisor desiring to add or cancel a plan, or change the existing title of a plan, must submit this change in writing, on the form illustrated in Figure 3. This form, properly filled out and signed, is automatically routed to the clerical section keeping the file records and where the plan numbers are assigned, and then to the scheduling group where the schedules are corrected accordingly. The change notices are filed for future reference, in case changes are ever questioned and in case it is ever necessary to trace the history of a plan.

Besides being used for production control the Plan Schedules are also used to estimate the monthly or total expenditure of manpower required to design a given project. From the way the charts are set up all that is necessary to determine the number of men required for each month is to count the number of the heavy horizontal lines within the shaded areas, on each schedule, and then add them up. By extending this method to all the schedules in use it is thus possible to estimate the workload for the Machinery Division over the whole period of time covered by these schedules.

All Schedules are drawn on tracing cloth and reproduced in blueprint form. Copies of the Schedules are sent to the Production Division, the Material Procurement Section and the other Design Sections. The Section Chiefs of the Machinery Division receive copies of the Plan Schedules applicable to their groups.

NAVY YARD, N. Y.
DESIGN SECT.—MACHY.

CHANGE OF DESIGN SCHEDULE NOTICE

Date.....

To: L21
L30
L-66 or Technical Library
L-33

PLAN NO. TITLE

APPLICABLE TO FOLLOWING SHIPS:

Estim. Starting Date.....

Estim. Completion Date.....

NATURE OF CHANGE:

( ) Change of title (avoid if possible)

Section.....

( ) Cancelled

( ) Addition (new plan) Supervisor.....

( ) Change in scheduled date—

Approved.....
Assistant Design Supt.

FIGURE 3.

The system of scheduling described here has been in use in the Machinery Division since 1938, when we were designing the then super dreadnaughts, the North Carolina class. Admiral J. J. Broshek, U.S.N., then Captain and Design Superintendent at the Navy Yard, was the first to realize the value of schedules in the drafting room. It was under his direction that the schedules were first started in the Machinery Division. The Master Erection Schedule and a corrected set of Plan Schedules were kept in his office in a special rack provided for this purpose. He made constant use of the schedules in following up the design work of the office.

Our experince with these schedules has been, to say the least, very satisfactory. Use of the plan schedules has resulted in a tremendous improvement in plan production. Even though our schedules are based mostly on estimates nevertheless, the results obtained are tangible and concrete.

LIGHT TRAPS.

By B. F. GREENBERG.*

INTRODUCTION.

The enemy should be able to see our ships at night. What he must see, however, is not lights but gun flashes. In order to make this possible, it is essential that darkened ship conditions be obtained.

"Darkened ship" can be defined as the condition whereby lights on a ship can not be seen beyond a specified safe limit. The safe limit for the purpose of this report is assumed to be 400 yards. Thus, when a vessel is operating under darkened ship conditions, the light from the ship on a dark night should not be visible at a distance of 400 yards. All the methods of obtaining light security, with the exception of the use of door switches, involve light barriers or traps. A light trap is an arrangement of panels which will trap direct light, and reduce the brightness of visible panels to a level which cannot be seen beyond a safe limit.

There is no one type of light trap which can be used for all applications. Each point of possible light exposure presents a different problem. There are, however, certain general rules which can be applied to all cases. By understanding these rules and the concepts underlying their derivation, it is possible to anticipate the best solution for each application.

It is the intent of this paper to outline step by step, the development of the general concepts and rules applying to the attainment of darkened ship conditions. Included in this report are: (1) a method of calculating light trap efficiency; (2) a graph for computing the spacing of fixtures from different types of light openings; and (3) a general method for calculating, under any set of conditions, the distance from which the lights of a ship can be seen.

GENERAL CONCEPTS.

As previously mentioned, a light trap is a means for reducing the amount of light visible in a given direction. For example,

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if 1000 units of light fall on one entrance of a light trap, then a fraction of this light, (of the order of 1 unit of light) is visible through the outer entrance. Now, if the permissible amount of spill light to the outside is equal to or less than 1 unit, then it follows that, in order to obtain light security with a light trap having an effectiveness of 1000 to 1, the amount of light falling on the inboard entrance should not exceed 1000 units. If the incident light were greater than 1000 units, the amount visible to the outside would be greater than one; consequently light security would be violated. However, if the incident light were less than 1000 units the minimum standard of light security would be exceeded.

The effectiveness of a light trap depends for the most part on the number of surfaces involved, the color of the paint, the size of visible opening, and the geometrical arrangement of panels. The method of calculating light trap effectiveness for any type of arrangement is discussed in Appendix A.

The receiver used in evaluating the effectiveness of a light trap is a very sensitive organ, the eye. The light of a match on a dark night can be seen for many miles. Consequently, in order to keep the visibility of a light source to a short distance, the amount of light emitted by the source must be kept to a minimum.

The amount of light which cannot be seen beyond a specified distance has been the subject of many tests. The results of the tests indicate that under any set of conditions, a certain amount of light is required at the eye of the observer in order that the source of the light can be ascertained. The amount of light required in order to just see the source is known as the threshold illumination.

The relation between the source of light and illumination for threshold conditions is similar to that in any illumination problem.

The most simple, and the most useful, illumination formula is the "inverse square law". According to this formula, the illumination from a source of candlepower I, at a distance of D from the source, is inversely proportional to the square of the distance between the source and the point, P, in question, and directly proportional to the intensity.

From this formula the maximum permissible candlepower can be calculated for a given threshold illumination and an allowable visibility distance.

Table I

Background Brightness	Threshold Illumination (Ft. Candles)					
	Incandescent Lamp			+Red Filter		
	Size Source			Size Source		
	1'	1°	10°	1'	1°	10°
Overcast Night 1×10^{-5} foot lamberts	2.8×10^{-10}	13×10^{-10}	190×10^{-10}	35×10^{-10}	175×10^{-10}	2500×10^{-10}
Clear Night 1×10^{-4} foot lamberts	3.2×10^{-10}	15×10^{-10}	220×10^{-1}	40×10^{-10}	200×10^{-10}	2900×10^{-10}
Moonlight 1×10^{-3} foot lamberts	3.9×10^{-10}	18×10^{-10}	270×10^{-10}	49×10^{-10}	245×10^{-10}	3500×10^{-10}

The threshold illumination, (i.e., the required amount of light at the eye of the observer in order that the source can be just seen) is dependent upon a number of variables. In fact there is a certain range of values for any set of conditions in which the light is sometimes seen and sometimes missed. Consequently, threshold conditions must be defined along with the ratio of the number of times the light is presented to the number of correct observations for the assumed threshold value. For the purposes of this paper a threshold of 50 per cent has been assumed. The threshold illumination in foot candles for different conditions of background brightness, size of test source, and colors of test source is shown in Table I. A more complete discussion on threshold illumination is given in Appendix B.

Although the threshold illumination depends upon a number of variables, for the purpose of this study a threshold value can be agreed upon by assuming the worst possible conditions from the point of view of being detected. From Table I, it will be noted that this condition occurs with an overcast night, and a point source. For this condition the threshold illumination is 2.8×10^{-10} foot candles.

Utilizing the inverse square law, and a threshold illumination of 2.8×10^{-10} foot candles, the maximum permissible intensity for a visibility distance of D is equal to:

$$I = Ep\ D^2 \qquad I = 2.8 \times 10^{-10}\ D^2 \qquad (I)$$

From the above formula it is evident that, in order to specify a permissible candlepower, criteria for specifying the allowable visibility distance must be agreed on. For the purposes of this study a value of 1200 feet is assumed. Substituting the value of D = 1200 feet in Equation I the premissible candlepower is:

$$I = 2.8 \times 10^{-10} \times (1200)^2 = 400 \text{ microcandles.}$$

The maximum allowable candlepower, for a visibility distance of 1200 feet on an overcast night is 400 microcandles. For different background conditions and visibility distances the maximum allowable candlepower can be calculated. Using this value, the next step is to calculate the required fixture spacing from the visible opening, in order that the candlepower visible to the outside does not exceed the permissible limit.

Fixture Spacing Required to Obtain Light Security (White Light).

The candlepower of a source of uniform brightness, when considered at a distance from the source greater than approximately ten times the largest dimension of the source, is equal to

$$I = \frac{BA}{\pi} \qquad (II)$$

Where

I = Candlepower,
B = Brightness (foot lamberts),
A = Area of the source (square feet).

The brightness, B, is a measure of the light coming from the surface. Illumination, E, is a measure of the light falling on a surface. The relation between illumination and brightness for a diffusing surface is given by the relation: For a diffuse surface the brightness is equal to the illumination on the surface times the reflection factor. Thus, equation II can be written as:

$$E = \frac{\pi I}{\rho A} \qquad (III)$$

Where

E = illumination in foot candles,
ρ = reflection factor.

The area, A, in the above equation applies to the area of the visible light source, which, in the case of shipboard application, is the area of the bulkhead opening. Assuming an area of 8 square feet, equation III becomes $E = \frac{.39I}{\rho}$. The candlepower, I, refers to the maximum permissible candlepower, which for a visibility of 1200 feet, and a background brightnesss equivalent to an overcast sky, is 400 microcandles. Substituting these values in equation III the permissible illumination, on a surface is:

$$E = \frac{.39I}{\rho} = \frac{.39 \times 400 \times 10^{-6}}{\rho} = \frac{156 \times 10^{-6}}{\rho} \text{ foot candles} \qquad (IV)$$

Thus, for reflection factors of 80, 50, 10, 1,01, and .001 per cent the permissible illuminations are 196×10^{-6}, 312×10^{-6}, 156×10^{-5}, 156×10^{-4}, and 156×10^{-3} foot candles respectively.

The illumination from a fixture at any point depends upon the total light emitted from the luminaire and the distribution of the light, as well as the distance and angle from the source to the point in question. The maximum illumination on a vertical plane, expressed in foot candles, at varying distances from different luminaires is given in Figure 1. The calculation of illumination values on the graph are based upon the illumination distribution of lighting fixtures used aboard ship. From this graph, the illumination on a vertical surface at distances of from 4 to 50 feet from reflector and globe type of lighting fixtures of varying wattage can be obtained. For example, (from Figure 1,) the illumination on a vertical surface 15 feet from a 50 watt globe type fixture is .16 foot candles, while the illumination 21 feet from a 100 watt reflector unit is .1 foot candle.

Equation IV expresses the permissible illumination. Figure 1, enables the determination of illumination, at any distance. Combining these two factors the required fixture spacing for a given reflection factor or the required reflection factor for a given spacing can be determined. For example, for a reflection factor of 2 per cent, the permissible illumination is 78×10^{-4}. Therefore, in order to obtain an illumination equal to or less than this value from a 50 watt reflector type fixture, the unit should be placed at a distance equal to or greater than 45 feet.

The reflection factor of a dark black paint is of the order of 2 to 3 per cent; therefore, according to the above calculations two or more surfaces must be used in order to obtain fixture separations less than fifty feet. If more than one surface is interposed between the source of light and the visible opening, the reflection factor of the combination is equal to the product of the reflection factors of each surface. For example, the combined reflection factor of two black surfaces, each with a reflection factor of 3 per cent, is $.03 \times .03 = 9 \times 10^{-4}$. Thus, with two surfaces painted black each with a reflection factor of 3 per cent, interposed between the source of light and the outside, the required fixture spacing of a 50 watt reflector type fixture to obtain the illumination of $E = \frac{156 \times 10^{-6}}{9 \times 10^{-4}} = .17$ is 12 feet. This

REQUIRED FIXTURE SPACING TO OBTAIN LIGHT SECURITY

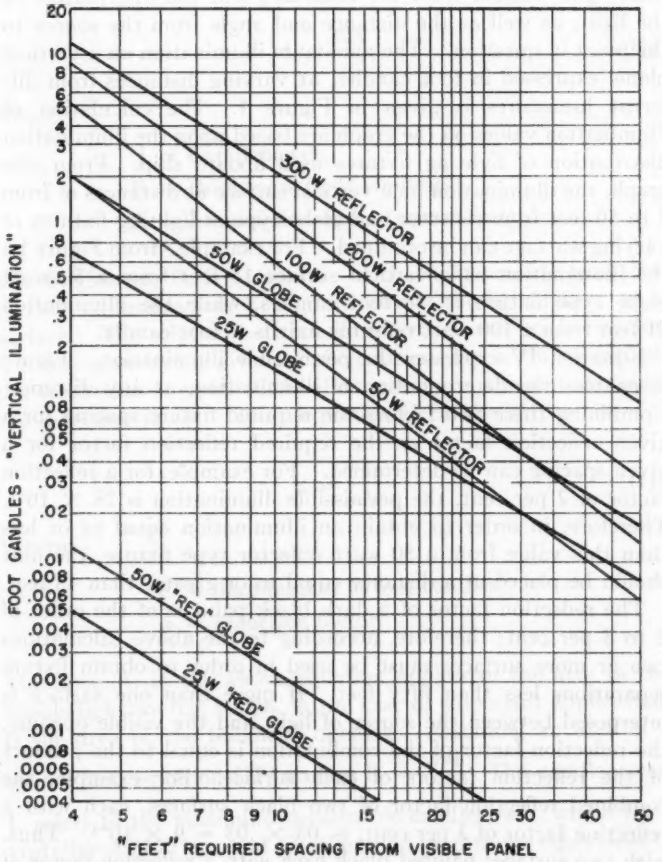


FIG. I

distance is measured from the source of light to the surface visible to the outside.

The following examples illustrate required fixture spacing calculations for different panel arrangements and reflection factors.

A. Find the required spacing of a single 50-watt reflector type fixture from an eight square foot panel visible outboard. The color of the visible panel is black, reflection factor 3 per cent, and a second panel painted gray, reflection factor 36 per cent, intercepts the light. The permissible visibility distance is 400 yards.

Solution: The permissible illumination calculated from equation III is equal to

$$E = \frac{\pi \times 400 \times 10^{-6}}{.03 \times .36 \times 8} = \frac{156 \times 10^{-6}}{.03 \times .36}$$

$$E = .014 \text{ foot candles.}$$

Now, again from Figure 1, the spacing of a 50 watt reflector type fixture from a panel so as to obtain an illumination of .014 foot candles is 36 feet.

B. Determine the required spacing of two 100 watt reflector type fixtures from a panel visible outboard. Each fixture should be spaced at equal distances from the visible panel. The area of the outboard opening is eight square feet. The visibility distance is 400 yards. The two black reflecting surfaces intercepting the light from the fixture have reflection factor of 3 per cent.

Solution. The permissible illumination on the outboard panel can be obtained by substituting in Equation IV, thus $E = \frac{156 \times 10^{-6}}{.03 \times .03} = .17$ foot candles. Since each of the two fixtures is equally spaced from the visible panel, the illumination from each is the same and equal to $\frac{.17}{2} = .085$ foot candles. Now, (from Figure 1) in order to obtain .085 foot candles the spacing of a 100 watt reflector type fixture is 23 feet.

C. Determine the maximum size reflector type fixture which could be placed at a distance of 25 feet from the eight square

foot visible opening. The reflection factor of each of the two intercepting panels is 3 per cent. The permissible visibility distance is 400 yards.

Solution: The permissible illumination is .17 foot candles. Since this is equal to the illumination from a 200 watt reflector type fixture at 24 feet, the permissible fixture size for a reflector type fixture is 200 watt or less.

D. Find the required spacing of the fourth fixture in a compartment where three reflecting surfaces intercept light visible outboard. The reflection factor of the three surfaces is 3 per cent, 3 per cent, and 20 per cent. One hundred watt reflector type fixtures are installed and each of the three fixtures are at a distance of 20 feet from the visible opening. The area of the visible opening is 8 square feet and the permissible visibility distance is 400 yards.

Solution. From Equation IV the permissible illumination falling on the visible panel can be calculated as equal to

$$E = \frac{156 \times 10^{-6}}{.03 \times .03 \times .20} = .85 \text{ foot candles.}$$

From Figure 1, the illumination from a 100 watt reflector type fixture at a distance of 20 feet from the fixture is .12 foot candles. The contribution of the three fixtures is then $3 \times .12 = .36$ foot candles. The permissible contribution of the fourth fixture is .85—.36 foot candles or .49 foot candles. From Figure 1, the required spacing of a 100 watt reflector type fixture so as to obtain .49 foot candles is 11 feet. Thus, with an arrangement of three reflecting surfaces, of reflection factors equal to 3 per cent, 3 per cent, and 20 per cent, a permissible arrangement of four 100 watt reflector type fixtures so as to obtain light security is 20, 20, 20 feet and 11 feet respectively.

Fixture Spacing Required to Obtain Light Security (Red Light).

In accordance with Table 1, and as described in Appendix B, the threshold illumination for red lights similar to those used aboard ship is the order of thirteen times that for white light. This means that in order to just see a red light the amount of light at the eye of the observer must be at least thirteen times that needed to just see a white light. Because more red light

than white light is required to just see a source of red light, the required spacing of a red light source from a panel visible outboard is not as great as that for white light.

The means for obtaining red light aboard ship consists, for the most part, of using incandescent lamp sources and placing a red filter over the source so as to filter out all light except red light. The per cent of red light to the total amount of light emitted by the incandescent lamp source is of the order of 13 per cent. The red filter placed over the source does little to change the illumination distribution. Thus distribution of illumination from a red light fixture is the same as that from a white light fixture except that the amount of light is 13 per cent of that of a similar white light fixture.

The illumination distribution curves for the red light fixtures as shown on Figure 1, are not the actual values, but rather illumination distribution related to the threshold of white light. It will be noted in Figure 1, the relative distribution of red light fixtures is $1/13 \times 1/13 = \frac{1}{169}$ that of the equivalent white light fixtures, since the threshold illumination of white light is 1/13 that of red light and the intensities when using a red filter is 1/13 that of the rated intensity of white light.

The method of calculating the required fixture spacing for red light fixtures is similar to that for white light fixtures. Utilizing Figure 1, it will be noted that for 400 yard visibility the required spacing, from a 50 watt red globe fixture, illuminating an eight square foot panel visible outboard, is as follows:

Color of Panel	Reflection Factor	Required Spacing
Black.....	3 per cent	7 feet
Gray.....	36 per cent	18 feet
White.....	70 per cent	25 feet

It has been previously shown that the permissible candlepower for 400 yards visibility is 400 microcandles. Now, since the threshold for red light is thirteen times that for white light, substituting in equation I, the maximum permissible red light candlepower is $13 \times 400 = 5200$ microcandles. Based on this

value the permissible illumination on a panel visible to the outside as obtained from Equation III:

$$E = \frac{\pi \times 5200 \times 10^{-6}}{\rho A} = \frac{16328 \times 10^{-6}}{\rho A} \quad (V)$$

According to the inverse square law the illumination at a point is directly proportional to the intensity and inversely proportional to the square of the distance between the source and the point in question. The intensity "I", in candles, of a 50 watt diffusing type of red light fixture, similar to that used aboard ship, is 3 candles. Assuming that "N" red light fixtures are employed, all equidistant from the visible panel, then the intensity of the combination is equal to "3N". Utilizing the inverse square law, and substituting the value for permissible illumination as given above and the intensity value for combinations of 50 watt red light fixtures, then the required fixture spacing from the visible panel can be calculated from the following formula.

$$E = \frac{16328 \times 10^{-6}}{\rho A} = \frac{I}{d^2} = \frac{3N}{d^2}$$

$$\frac{16328 \times 10^{-6}}{\rho A} = \frac{3N}{d^2}; \qquad d^2 = \frac{3 \times N \times \rho \times A}{16328 \times 10^{-6}} \quad (VI)$$

$$d = 13.6 \sqrt{A \times \rho \times N}$$

Where

A = Area of the visible opening (square feet),

ρ = Reflection factor of the surface visible to the outside,

N = Number of red light fixtures.

For example, if the area of a bulkhead opening is assumed to be eight square feet, and the reflection factor of black paint is .03, thus by substituting these values in equation VI, the required spacing of one red light fixture so as to insure light security is

$$d = 13.6 \sqrt{8 \times .03 \times 1} = 6.7 \text{ feet.}$$

Assuming two equally spaced red light fixtures are employed then the required spacing of each fixture from the visible panel is

$$d = 13.6 \sqrt{8 \times .03 \times 2} = 9.4 \text{ feet.}$$

Light Trap Design.

Up to this point a mathematical basis for light trap design was discussed and methods for calculating the required fixture spacing reviewed. Based on the results of these calculations certain general rules related to light trap design can be formulated.

From Figure 1 it will be noted that using white light, the required fixture spacing from a single black panel, of eight square feet in area, is a distance greater than 45 feet. Consequently, for white light fixtures it can be stated that:

> In no case shall the source of light be visible to the outside;
>
> At least two reflecting surfaces shall intercept the light visible to the outside;
>
> A line projected from the source to the outside should intercept two surfaces.

Again from Figure 1, it will be noted that in order to limit the required white light fixture spacing to less than 40 feet, the combined reflection factor of the surfaces should be less than 3 per cent. Consequently, it can be stated that:

> In all cases at least one surface shall be painted with a dull black paint.

It has been demonstrated that the ability of a surface to trap light is dependent upon the reflection factor of the surface; therefore, it can be stated that, for optimum trap performance:

> Each reflecting surface, and adjoining surfaces shall be painted with dull black paint.

Light Trap Constants.

The most potent tools in light trap design are the number of light intercepting panels and the reflection factor of each. By methods already explained, the required fixture spacing can be calculated. The resulting calculations are based on the light

reflecting surface being of a specular nature; that is, the surfaces are mirror-like and reflect the incident light in a direction dependent upon the position of the light source.

A second type of surface is known as a diffuse type, more commonly thought of as having a flat finish. Light striking a diffuse surface is reflected back in all directions. Because of the dispersion of light by diffuse surfaces, the passage of light through an arrangement of diffuse surfaces is less efficient and consequently, is of greater value in light trap design.

The effectiveness of an arrangement of surfaces to trap light can be expressed in terms of a factor known as a "light trap constant." By definition, this factor can be considered independently of the reflection factors, and expressed as the ratio of the amount of light at the outer surface to the amount of light at the inner surface. Light trap constants for different types of light traps are shown in Figures 2-6 inclusive. The method

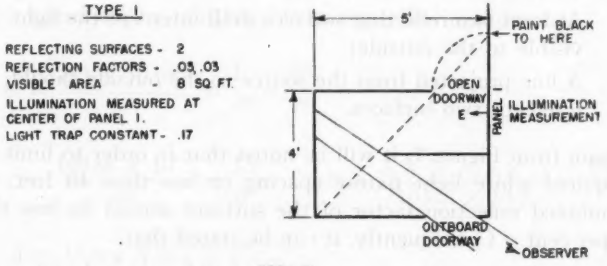


FIG. 2

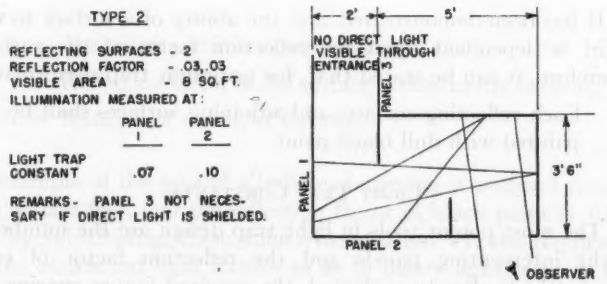


FIG. 3

TYPE 3

	PAN. 1	PAN. 2
ILLUMINATION	E_1	E_2
REFLECTING SURFACES	3	2
LIGHT TRAP CONSTANT	.07	.26

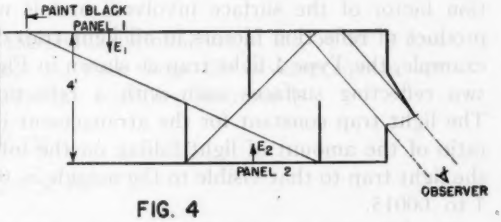


FIG. 4

TYPE 4

REFLECTING SURFACES - 3
ILLUMINATION - PANEL 1
LIGHT TRAP CONSTANT - .02

REMARKS - NO DIRECT LIGHT ZONE A

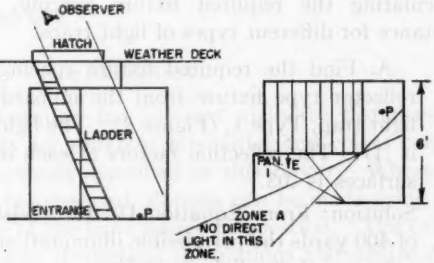


FIG. 5

TYPE 5

REFLECTING SURFACES - 2
ILLUMINATION - FLOOR BASE OF LADDER
LIGHT TRAP CONSTANT - .03

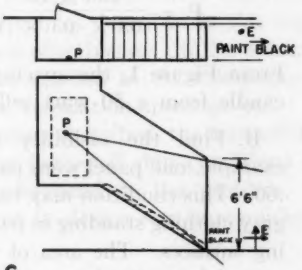


FIG. 6

of calculating the different light trap constants is described in Appendix A.

The light trap constant is another factor in reducing the light visible to the outside. In effect the light trap constant which is always less than unity, can be considered as reducing the reflection factor of the surface involved, and is multiplied by the product of reflection factors in all light trap calculations. For example, the Type 1 light trap as shown in Figure 2, consists of two reflecting surfaces each with a reflection factor of .03. The light trap constant for the arrangement is .17. Thus, the ratio of the amount of light falling on the inboard entrance of the light trap to that visible to the outside is .03 × .03 × .17 or 1 to .00015.

The following examples will serve to illustrate the method of calculating the required fixture spacing, and the visibility distance for different types of light traps.

A. Find the required fixture spacing of a 50 watt reflector type fixture from the inboard entrance of a light trap, Type 1, (Figure 2). The light trap constant is .17. The reflection factors of each of the reflecting surfaces is .03.

Solution: From Equation IV, for a visibility distance of 400 yards the permissible illumination of the inboard entrance of the light trap is

$$E = \frac{156 \times 10^{-6}}{.03 \times .03 \times .17} = 1.0 \text{ foot candle.}$$

From Figure 1, the spacing required to obtain 1 foot candle from a 50 watt reflector type fixture is 6 feet.

B. Find the visibility distance, if in the above example, one panel were painted gray, reflection factor .60. This condition may be simulated by a person with gray clothing standing in front of one of the light reflecting surfaces. The area of the visible opening is eight square feet. Neglect atmospheric attenuation.

Solution: From Equation III, the effective candlepower visible to the outside is equal to

$$I = \frac{E_p \rho_1 \rho_2 k A}{\pi}$$

$$I = \frac{1 \times .03 \times .60 \times .17 \times 8}{\pi} = .0078 \text{ candles.}$$

Where

k = light trap constant.

Assuming a threshold illumination of 2.8×10^{-10} foot candles (Table 1), then from the inverse square law (Equation III), the visibility distance is

$$I = E D^2$$

$$.0078 = 2.8 \times 10^{-10} D^2$$

$$D = \sqrt{78 \times 10^{-4}/2.8 \times 10^{-10}} = 5.28 \times 10^3$$

$$D = 5280 \text{ feet.}$$

Conclusion.

The intensities required for different visibility distances as well as the visibility for different intensities and colors can be obtained by the methods described in this report. Where an extended source is involved calculations can be made using the brightness and area of the light source in question. The calculations are based upon theoretical calculations and utilize constants obtained by tests. Calculations utilizing the formula described in this report are in good agreement with test results.

Utilizing the formulae and principles described in this report a method of evaluating light trap effectiveness can be obtained. In addition these factors can be applied to the calculation of visibility distances from airports, ventilation openings, etc.

By an understanding of the factors described, the best solution to the many questions involving darkened ship conditions can be obtained. Knowing the background to the problem, the myriad of details and exceptions in design can be confidently handled.

Appendix A

Calculation of Light Trap Constants.

There are many possible types of light traps and many variables in each type of arrangement. In order to predetermine the performance of each type of light trap, a mathematical

procedure has been designed. The proposed method involves the calculation of a "light trap constant." The "light trap constant" can be defined as the ratio of the amount of light at the outer surface to the amount of light at the inner surface, assuming unity reflection factor at each surface.

The illumination at a point on a panel parellel to another panel of uniform brightness can be calculated from the formula:

$$E_2 = B_1 k_z \qquad \text{(VII)}$$

where "k_z" is a factor depending upon the size and separation of the panels and is equal to

$$k_z = \frac{1}{2\pi}\left[\frac{H}{\sqrt{D^2 + H^2}}\sin^{-1}\frac{W}{\sqrt{H^2 + W^2 + D^2}} + \frac{W}{\sqrt{D^2 + W^2}}\sin^{-1}\frac{H}{\sqrt{H^2 + W^2 + D^2}}\right]$$

B_1 = brightness (foot lamberts) of panel 1.

E_2 = illumination of panel 2 (foot candles).

The average brightness of panel 2 resulting from a uniform brightness of panel 1 is equal to

$$B_2 = E_2 \rho_2 = k_z B_1 \rho_2, \text{ as } B_1 = \rho_1 E_1 \qquad \text{(VIII)}$$
$$B_2 = k_z E_1 \rho_1 \rho_2$$

Where

B_2 = brightness of panel 2,

k_z = light trap factor where two parallel surfaces are involved,

$\rho_1 \rho_2$ = reflection factor of surfaces 1 and 2, respectively,

E_1 = average illumination on surface 1.

Similarly, the illumination at a point on a panel perpendicular to another panel of uniform brightness can be calculated from

the equation:

$$E = B\,k_y \qquad \text{(IX)}$$

Where k_y = factor relating the size of the panel and is equal to

$$k_y = \frac{1}{2\pi}\left[\tan^{-1}\frac{W}{D} - \frac{D}{\sqrt{D^2+H^2}}\sin^{-1}\frac{W}{\sqrt{D^2+H^2+W^2}}\right]$$

The average brightness of panel 2, resulting from a uniform brightness of panel 1 perpendicular to panel 2 is equal to

$$B_2 = k_y\,E_1\,\rho_1\,\rho_2$$

Where

E_1 = average illumination of panel 1 (foot candles),

$\rho_1\,\rho_2$ = reflection factors panels 1 and 2 respectively,

B_2 = brightness panel 2 (foot lamberts),

k_y = light trap factor when two perpendicular surfaces are involved.

Utilizing the above procedure the method of computing the light trap constant for the light trap Type 1, can be obtained as follows:

(a) Assuming a uniform illumination on panel 1, the brightness of panel 1, can be calculated from the formula:

$$B_1 = \rho_1\,E_1$$

(b) The illumination at any point on panel 2 resulting from the uniform brightness of panel 1 can be obtained from equation VII.

For example, the illumination at point "a" equals:

Where

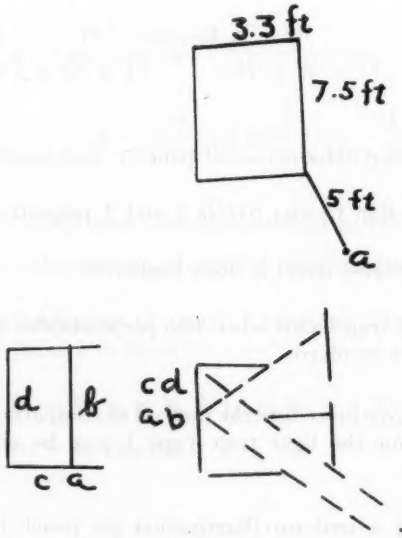
H = 7.5 ft. D = 5 ft. W = 3.3 ft.

$$E_a = .13\ B$$

Similarly, the illumination at points b, c, d, equal:

$$E_b = .21\ B_1 \qquad E_c = .12\ B_1 \qquad E_d = .19\ B_1$$

$$E_{2v} = .17\ B_1$$



The average brightness of panel 2 is then equal to

$$B_2 = \rho_2 E_2\,(av) = .17\ B_1 \rho_1 = .17\ \rho_1 \rho_2 E_1$$

Light trap factor equals .17.

For a non-uniform brightness of panel 1 the illumination at any point on panel 2 can be calculated by considering panel 1 as

made up of a number of small panels with a uniform brightness. The illumination at any point on panel 2 is then the summation of the product of the average brightness of each area and the factor $\frac{(E)}{B}$.

Appendix B.

Discussion of Threshold Illumination.

The ability to perceive the presence of light depends upon the illumination from the source of light in question at the eye of the observer. The required illumination just to see a light source is a function of the size, position, color and background of the light source. The average values for different conditions are given in Table I of the text. The factors involved in the determination of threshold values have been studied by many investigators, and from a review of the many papers on the subject the threshold values for any set of conditions can be obtained.

Threshold Range.

Illumination units are based upon the physiological characteristics of a so-called standard observer. The characteristics of the eye of any individual observer may vary from standard conditions. Consequently, it is not possible to state a precise value of threshold illumination above which the light can be seen, and below which it can not be seen. In reality there is

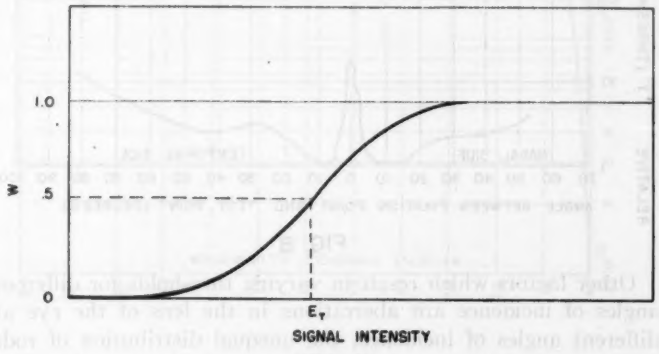


FIG. 7

always a certain range of intensities within which the light is sometimes seen, sometimes missed. If the ratio of the number of times a light source is seen to the number of times it is presented is plotted against the illumination at the eye, a curve similar to that shown in Figure 7 is obtained. The threshold values for the purpose of this report can be taken as the intensity at which there is a 50 per cent chance of seeing the light source.

Position of Test Source.

The threshold illumination for any color is dependent upon the angle which the test spot subtends at the eye of the observer. The relative thresholds for white light at different angles is given in Figure 8. (The curve for any other colored light would be different.) From Figure 8 it will be seen that the relative thresholds at $\theta = 0$ degree and $\theta = 5$ degree differ by a factor 20. The reason for this large difference is that light falling upon the eye at an angle of 0 degree, falls on a portion of the retina in which there is a concentration of cones, while other portions of the retina contain mostly rods. The rods are more sensitive than the cones and are therefore used for night vision, i.e., illuminations levels of the order of .001 foot candles or less. The structure of the rods and cones differ, the rods being the coarser of the two. The adaption time for rod vision is of the order of 20-30 minutes, depending upon pre-adaption conditions, while the adaption time for cone vision is of the order of 2-3 minutes.

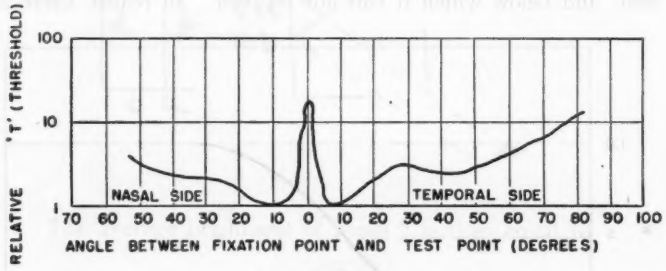

RELATIVE 'I' (THRESHOLD)
100
10
1
NASAL SIDE
TEMPORAL SIDE
70 60 50 40 30 20 10 0 10 20 30 40 50 60 70 80 90 100
ANGLE BETWEEN FIXATION POINT AND TEST POINT (DEGREES)

FIG. 8

Other factors which result in varying thresholds for different angles of incidence are aberrations in the lens of the eye at different angles of incidence, the unequal distribution of rods over the area of the retina and the spectral sensitivity of the rods.

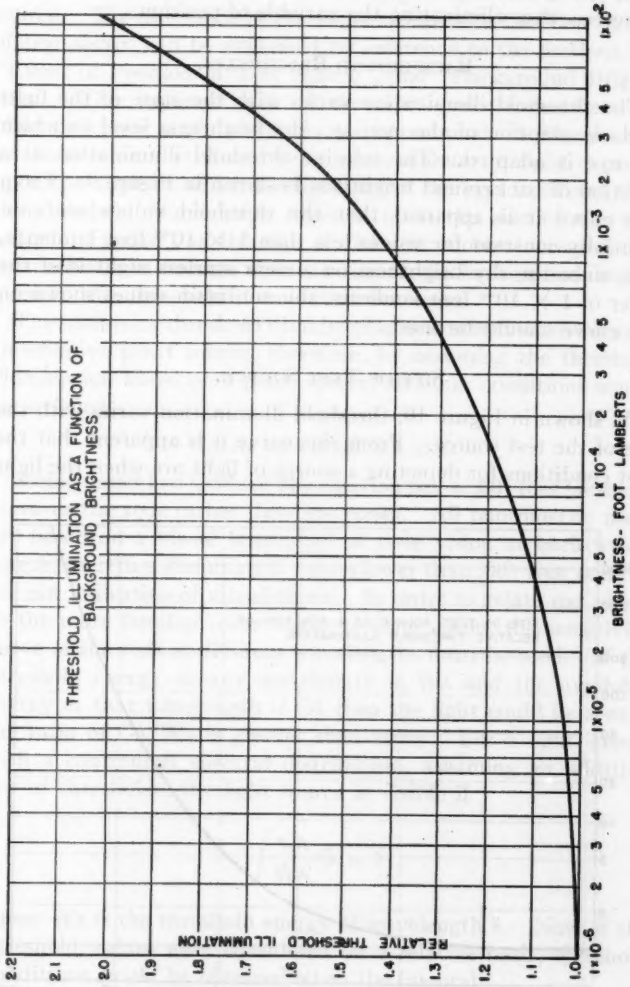

THRESHOLD ILLUMINATION AS A FUNCTION OF BACKGROUND BRIGHTNESS
RELATIVE THRESHOLD ILLUMINATION
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
1 X 10-6
2 3 4 5
1 X 10-5
2 3 4 5
1 X 10-4
2 3 4 5
1 X 10-3
2 3 4 5
1 X 10-2
BRIGHTNESS - FOOT LAMBERTS

FIG. 9

In the determination of threshold values it can be assumed that the observer would be searching the horizon and the light source would be picked up at the corner of the eye at an angle of 5 degrees, thus eliminating the variable of position.

Background Brightness.

The threshold illumination varies with the state of the light or dark adaption of the eye, i.e., the brightness level to which the eye is adapted. The relative threshold illumination as a function of background brightness is shown in Figure 9. From this curve it is apparent that the threshold values are substantially constant for values less than 1×10^{-4} foot lamberts, and, since the sky brightness on a clear starless night is of the order of 1×10^{-5} foot lamberts, the minimum values shown on the curve should be used.

Size of Test Source.

As shown in Figure 10, threshold illumination varies with the size of the test source. From this curve it is apparent that the best conditions for detecting a source of light are when the light

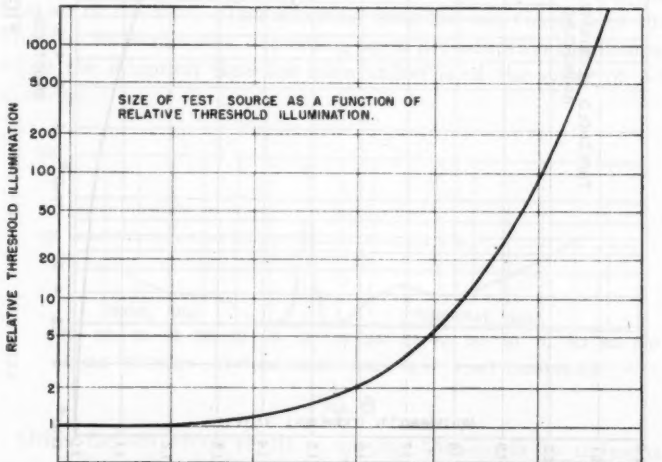


FIG. 10

can be considered as a point source. A source of light can be considered as a point source if the distance from which the light is viewed is many times greater the largest diameter of the light source. The difference in threshold values as a function of size of test source can be explained by reference to the sections on: "Effect of Position of Test Source", and "Background Brightness". The apparent size of the test source increases as a function of the angle which the source subtends at the eye, and as the sensation produced by a given amount of light depends upon the angle at which the light enters the eye, it follows that the threshold illumination depends upon the size of the test source. Further, as the size of the test source increases, the source tends to become equal in size to the background, consequently, the threshold illumination increases.

The minimum threshold illumination is obtained when viewing an effective point source; therefore, by assuming the threshold illumination based on a point source, all other conditions would be satisfied.

Color of Light Source.

Seeing under low levels of illumination below .001 foot lamberts involves the rods rather than the cones. All photometric units are based on a visual sensitivity of cone vision at each wavelength, therefore illumination values lower than .001 foot candles are not indicative of visual effect. In order to relate rod vision to the more familiar units based on cone vision, the sensitivity curve of the rods at different wavelengths must be used. If the threshold energy at any wavelength is $W\lambda$ and the available energy at that wavelength is $G\lambda$ then the light could be seen if the ratio of $G\lambda/W\lambda$ is greater than unity. For a light source with a continuous spectral distribution, assuming an additive law of thresholds, the light source is visible if

$$\int \frac{G\lambda}{W\lambda} d\lambda = 1$$

where $W\lambda$ is the threshold energy of wavelength λ. Now, if the threshold values were considered on a relative basis, threshold conditions would be obtained when the integral

$$\int \frac{G\lambda}{W\lambda} d\lambda$$

equaled a constant value. The threshold illumination for different spectral distributions can be obtained by relating the ratios of the summation of the product of the energy at each wavelength and the relative visibility of both the rods and cones. Thus, the threshold when using rod vision expressed in units based upon the cone visibility function is equal to

$$E_t = C_r \frac{\int_{400}^{700} G\lambda \, V\lambda \, d\lambda}{\int_{400}^{700} G\lambda \, U\lambda \, d\lambda}$$

Where

E_t = threshold illumination,

C_r = constant,

$G\lambda$ = energy distribution light source,

$V\lambda$ = relative visibility at different wavelengths cone vision (standard visibility function),

$U\lambda$ = relative visibility at different wavelengths rod vision.

From the above equation the threshold illumination in foot candles for any spectral distribution or color of test source can be calculated. For example, based on a threshold of illumination of white light as 2.8×10^{-10} foot candles, the order of magnitude of the threshold illumination for typical colored lights are:

Color	Threshold Illumination
White	2.8×10^{-10} f.c.
Blue	$.25 \times 10^{-10}$ f.c.
Green	1.1×10^{-10} f.c.
Orange	1.3×10^{-10} f.c.
Red	$35. \times 10^{-10}$ f.c.

In the above example it should be noted that the values are only typical and exact values can not be stated unless the spectral distribution of the colored light is known.

Other Factors.

There are several other factors influencing threshold illumination, namely: chromatic thresholds, effect of intermittent light, etc. The first of these factors involves color discrimination, and as the rods are not very sensitive to color, the threshold illumination required to distinguish differences in color is greater than the illumination required to just see the light source.

The second of these factors, namely, the effect of intermittent light, is a function of the total light reaching the eye in a given time. For intermittent light flashes, it can be assumed that the effective illumination is equal to a percentage of that of a steady source.

BIBLIOGRAPHY

(a) P. Moon: *Scientific Basis of Illuminating Engineering*, McGraw Hill, New York 1936.

(b) Stiles, Bennett and Green, Visibility of Light Signals with Special Reference to Aviation Lights, Report No. 1793, His Majesty's Stationary Office, London; 1937.

(c) B. J. Bouma: Illumination and Blackouts, *Phillips Technical Review* Vol. 4, No. 1, Jan. 1939.

PACIFIC OCEAN SURGE IN LOS ANGELES-LONG BEACH HARBOR.

By Donald L. Herr, Lieut. Commander, U.S.N.R., Member*.

I. Introduction.

The unusual Pacific Ocean wave phenomena experienced from time to time along the Southern California coast have been a subject of much study by scientists and engineers for a considerable period. One of the investigators has aptly remarked elsewhere[1] that "like all phenomena of nature not well understood, there are nearly as many theories of the cause of these manifestations as there are interviews given to the press whenever a spell of this type occurs which is sufficiently severe to become news." It is not a particular purpose of this article to create news; nor is it desired to add further to the confusion by propounding additional theories on the subject. This paper is intended as a record of certain physical facts considered to be new contributions of evidence leading to a better understanding of the complex water-wave phenomena encountered on the Pacific coast of Southern California, in general, and in Los Angeles-Long Beach harbor, in particular.

Depending upon the particular reason for interest in these phenomena by the various writers, the contemporary bibliography on the subject stresses one or more of the following aspects of the Pacific Ocean wave phenomena considered herein. They are called surges, seiches, ground swells, heavy swells, and "rip tides." They are referred to by their effects upon shore structures, such as pounding damage to berths, drydocks and piers in the harbor, breaching of shoreline bulkheads, disintegration of jetties, moles and breakwaters, undermining of water-front establishments, and severe erosion or piling-up of beaches. They are also referred to by their effects upon shipping, such as ships rolling heavily at their berths, "charging" alongside piers, and sickening rolling and pitching of ships anchored in the free ex-

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1. For all numbered references, see bibliography at end of paper.

panse of the harbor, or just outside. Many references may also be found to the loss of human life of bathers, fishermen, etc., who have been caught by the sudden appearance of the waves of large amplitude associated with the more intense manifestations of these peculiar Pacific Ocean wave phenomena. The files of the newspapers of Los Angeles, Long Beach and adjacent communities contain many written and photographic records of such manifestations.

Mr. Harry Leypoldt has elsewhere[1] given a clear description of the above-mentioned effects on shore-line and harbor structures, some of which have required extensive repairs and construction by the U. S. Army Engineers and local civil governments to remedy the situation. Others,[2] since long ago, have been keen observers of the effects of these peculiar waves in inducing severe rolling and pitching in vessels berthed or anchored in Los Angeles-Long Beach Harbor, and of the progressive effects of breakwater and jetty construction as added in the same area.

The reason for intense Naval interest in this phenomenon in the Los Angeles-Long Beach harbor area arises from the extreme damage experienced by pounding of ships against piers, difficult ship handling conditions and pounding of drydock caissons during drydocking operations, and the difficulty or impossibility of performing much work upon the ships which is normally capable of accomplishment when water-borne. It was in an effort to assist in the determination of the cause, nature and cure of this damaging wave action that the investigation reported upon was instigated.

Most scientific and engineering observers to date have considered the foregoing wave-action and effects peculiar to the Los Angeles-Long Beach Harbor to be the direct result of a phenomenon called by the mysterious word "seiche" (pronounced "saysh" in the French), first employed years ago by Dr. Forel[3] in describing the results of his investigations of certain types of water-waves which he encountered on Lake Geneva, Switzerland. "Seiche", as used by Dr. Forel and as defined by dictionary, refers specifically to a certain type of water-wave phenomenon, in which the waves *do not travel progressively* across the water expanse but, rather, consist of oscillations of the water-surface in which the positions of the "loops" (alternately crests and troughs in time) and of the "nodes" (points midway between those at

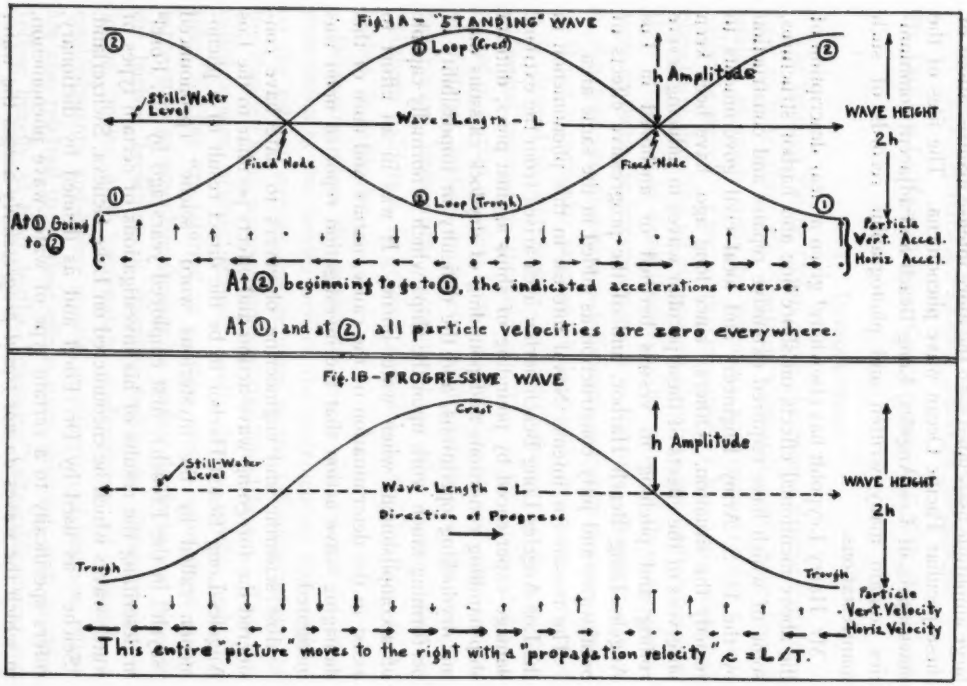


Fig. 1A – "STANDING" WAVE
② Loop (Crest)
h Amplitude
Still-Water Level
Wave-Length – L
WAVE HEIGHT 2h
Fixed Node
Fixed Node
① Loop (Trough)
At ① Going to ②
Particle Vert. Accel.
Horiz. Accel.
At ②, beginning to go to ①, the indicated accelerations reverse.
At ① and at ②, all particle velocities are zero everywhere.
Fig. 1B – PROGRESSIVE WAVE
Crest
h Amplitude
Still-Water Level
Wave-Length – L
WAVE HEIGHT 2h
Direction of Progress
Trough
Trough
Particle -Vert. Velocity
Horiz. Velocity
This entire "picture" moves to the right with a "propagation velocity" c = L/T.

which crests and troughs occur) are fixed in space. Such waves are called "standing waves" and, hence, "seiche" is identified with a certain type of standing waves.[4, 5, 6] (See Figure 1A.) "Seiche" may be physically identified with the type of water-waves observed in a rectangular pan of water after the tray has been suddenly tilted and then immediately restored to its original position; or, just as well, with some of the waves observed in a practically rectangular bath-tub after the whole mass of water has been set in motion by several strong sweeps of the hand. Most water-wave considerations of the Los Angeles-Long Beach Harbor phenomenon have been based upon this "seiche" concept.

The other water-waves commonly encountered in nature are the so-called "progressive" waves, in which the oscillations of the surface of the water-expanse consist of a "train" or continuous group of alternating wave crests and troughs which travel at a uniform rate of speed, with spacing between crests and troughs remaining essentially unchanged, across the open expanse of water. In other words, by singling out a particular crest or trough and following its movement, it is observed to move in a horizontal plane with a definite velocity of progress across the water surface. Such traveling water waves may be easily observed in the system of progressive waves created by dropping a stone or pebble into a body of water whose surface is otherwise smooth and ripple-free (See Figure 1B).

It is believed of salient engineering interest[7] that the results of the hydrodynamic investigation of San Pedro Bay (Los Angeles-Long Beach Harbor) reported herein give strong evidence that, whereas multitudinuous seiches or "standing waves", of the types described by Leypoldt[1] and others[2, 3, 4, 5, 6] no doubt exist and are excited from time to time in the Los Angeles-Long Beach Harbor, it is a fairly definite system of *progressive* Pacific Ocean waves which is responsible for the greater part of the damaging and hampering wave-action experienced at times by shipping and water-front construction[7] in this harbor. This system of progressive waves will be identified herein by the words "Pacific Ocean surge" or, simply, "surge", for want of a better name.

II. Los Angeles-Long Beach Harbor and Instrument Locations.

The particular region of interest is depicted, not to scale, in Figure 2. Los Angeles-Long Beach Harbor is bounded by San

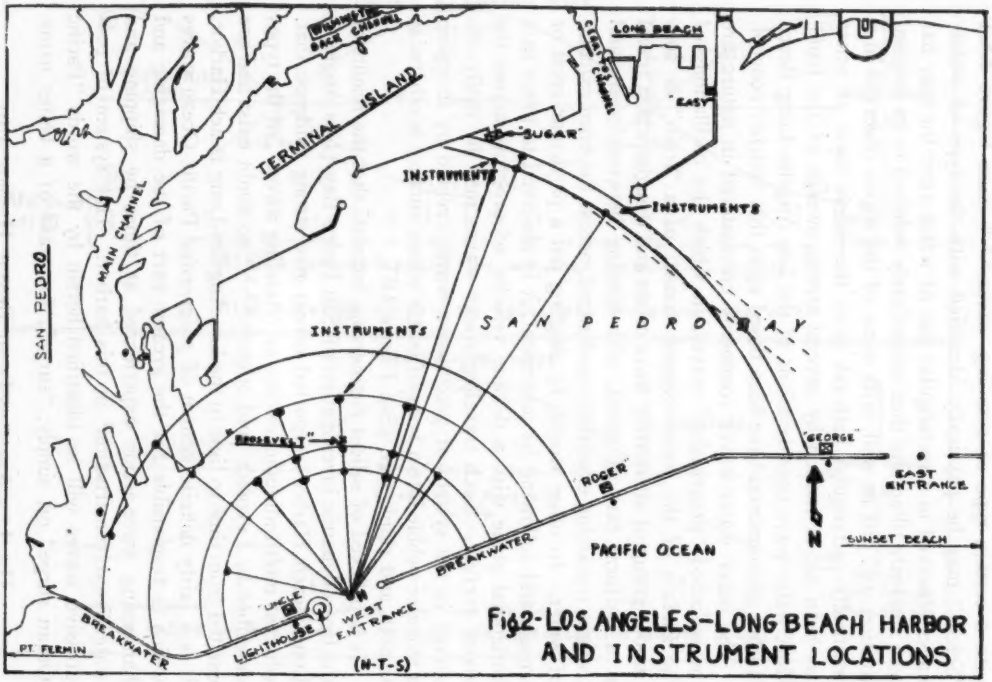


Fig2-LOS ANGELES-LONG BEACH HARBOR AND INSTRUMENT LOCATIONS

Pedro on the west, Long Beach on the east, Wilmington on the north and the Outer Breakwater on the south, towards the open ocean. Terminal Island is situated between San Pedro on the west and Long Beach on the east and is entirely surrounded by navigable waterways. The southern shore of Terminal Island faces the Pacific Ocean directly, except for the intervening Outer Breakwater. The Outer Breakwater itself runs from Point Fermin on the west to a point directly off-shore from the eastern half of Long Beach, and contains a "West Entrance" and an "East Entrance" for navigation into and out of the harbor. The open east end of the harbor is effectively sealed off to navigation by the curving shore-line and shoaling of the bottom. As may be seen in Figure 2, there are numerous slips, side-channels and sloughs in the area. The harbor may be considered to comprise an irregular area, roughly, seven miles long by three miles wide and, excepting flats and shoals, approximately forty feet in water depth.

The outer breakwater, built in portions over the past years[2], and consisting of a western, central and an eastern section, is, in general, of the semi-permeable type.

Three of the principal control and recording stations of this investigation were located at the points marked "Uncle", "Roger" and "George" on Figure 2, just inside this breakwater. The two other "permanent" stations were located at the points marked "Sugar" and "Easy" on Figure 2; "Sugar" was the central radio and operational control point for all operations. It will be noted that the first letters of the names given to the five control stations, "Sugar", "Uncle", "Roger", "George" and "Easy" spell out the word "S-U-R-G-E".

In addition to the 2 to 7 instrument-*recording* positions situated in each of these five stations, continuous data were also obtained for the point labeled "H" in Figure 2, in the middle of the West Entrance. The instrument-*measuring* positions, connected to these recording positions, were located in various distributions over the harbor area *and at various points outside the breakwater.* Certain additional data on the surge-waves in the harbor were collected at a subsequent control station, marked "Roosevelt" in Figure 2, from the cluster of points shown grouped around station "Roosevelt". Miscellaneous short-run data were collected at other points.

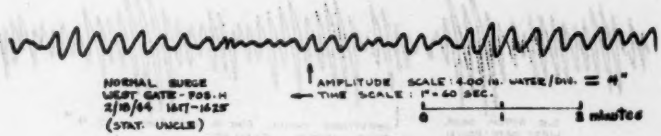
III. Types of Data and Measurements.

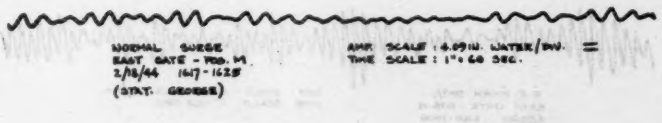
Over a period of six months, approximately 50,000 feet of recorded wave-data were obtained from the "S-U-R-G-E" and "Roosevelt" arrays of stations. These data consist of continuous, synchronized wave-measurements and recordings in which tidal and other long-wave effects were eliminated by the instruments. The instruments employed were photo-electronic and electronic in nature, converting, by these means, instantaneous voltages proportional to instantaneous pressure heights into calibrated, continuous deflections of a recording stylus across a moving recording tape. For the wave-lengths of interest it is readily shown that the accuracy obtained in utilizing instantaneous pressure heights to measure wave heights is of good engineering quality (normally better than 90% accurate). Typical such recordings of "normal" and "southeast storm" surge at various points over the harbor area are shown, respectively, in Figures 3 and 4, photo reduced by a factor of two. The time-scale and various amplitude-scales are as shown.

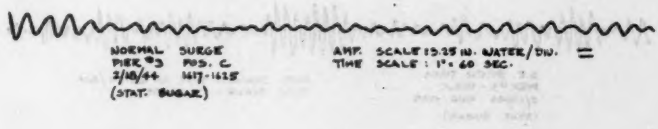
Over the immediately following period of two months, approximately 2500 feet of recorded wave-data were obtained from Stations "Sugar" and "Uncle"; these data consisted of continuous recorded periods during which tidal and all other long-waves were measured and recorded along with all waves of lesser wavelength. For this reason, these measurement-recordings will be identified as precision data. Typical such recordings of "quiet" and "intense" normal surge at Terminal Island, surge at the West Entrance, tidal and long waves, and a recording of a singular, approximately 6000-foot wave, obtained in this set of data, are shown, respectively, in Figures 5, 6, 7, 8, 9 and 10. The various time-and amplitude-scales of recording are as shown.

None of this vast body of data was obtained with the usual tide-gauge and water-stage recorders, for various reasons of precision and flexibility. However, the conventional tide-gauge and water-stage recordings were available for comparison and a typical one of each such type of recording is shown in Figures 11 and 12, respectively. It is to be noted that Figure 11 is typical of the best recording (i.e., fastest tape-speed and greatest amplitude-scale), made by a modified and speeded-up tide-gauge.

Early in the "S-U-R-G-E" measurement-recordings, it was quite apparent that ships berthed alongside the piers were closely







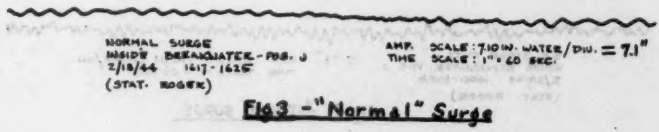
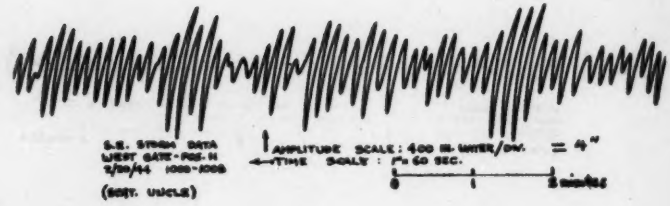
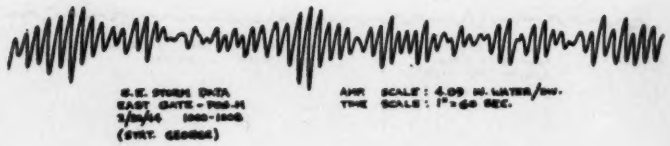
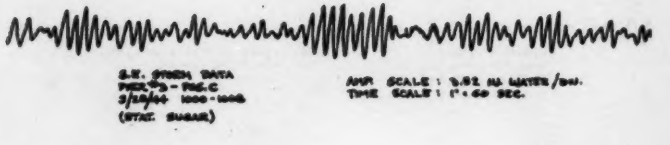


Fig 3 - "Normal" Surge







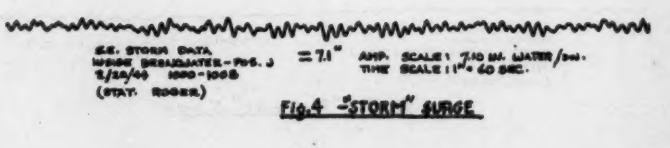
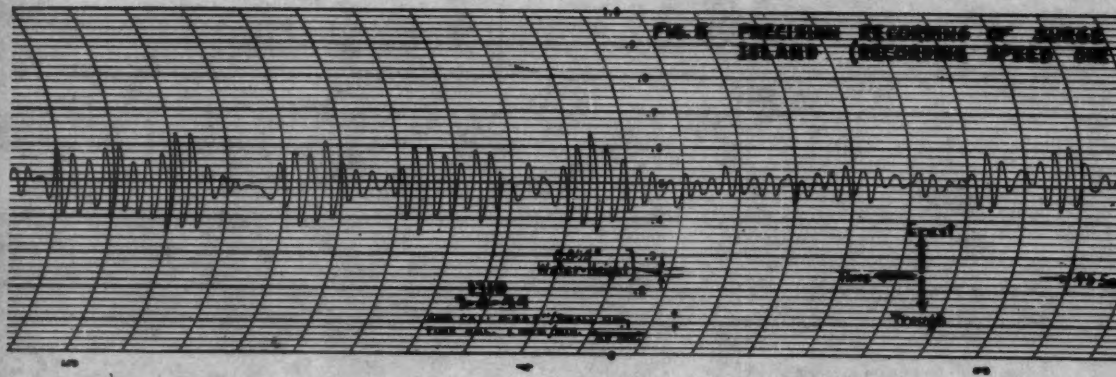
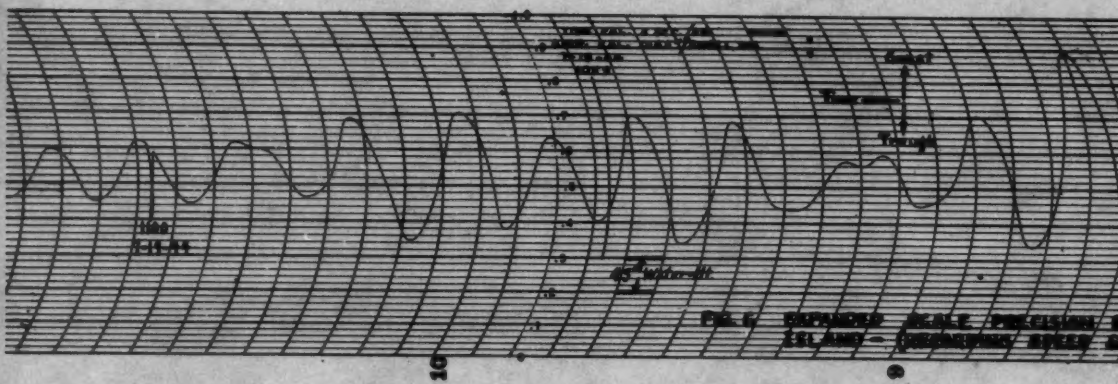
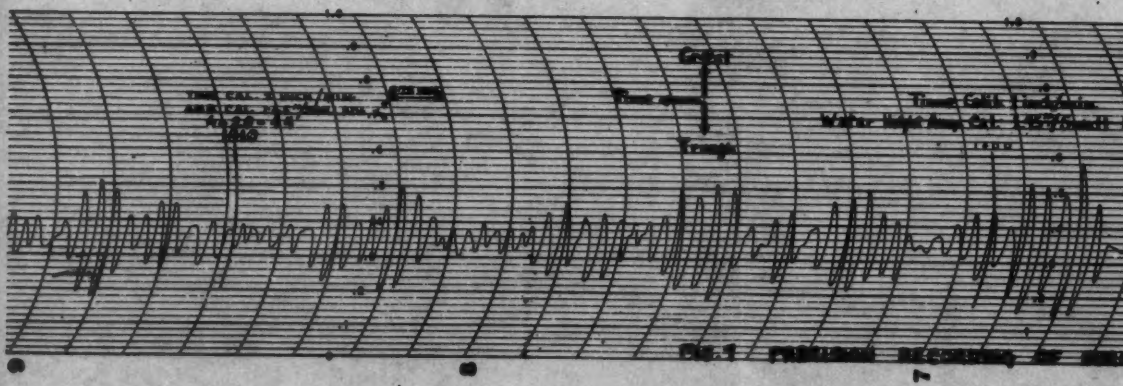
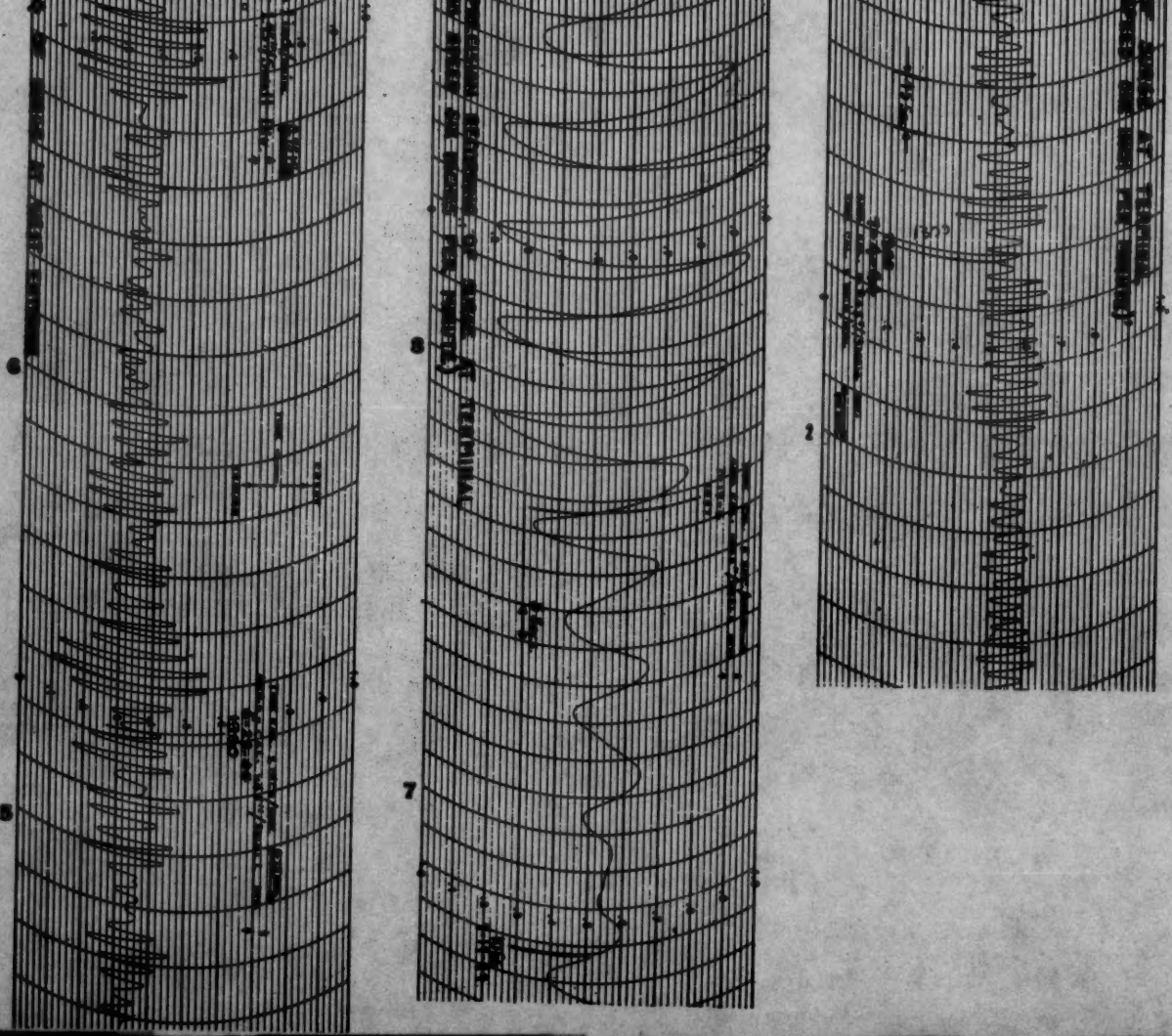


Fig.4 "STORM" SURGE









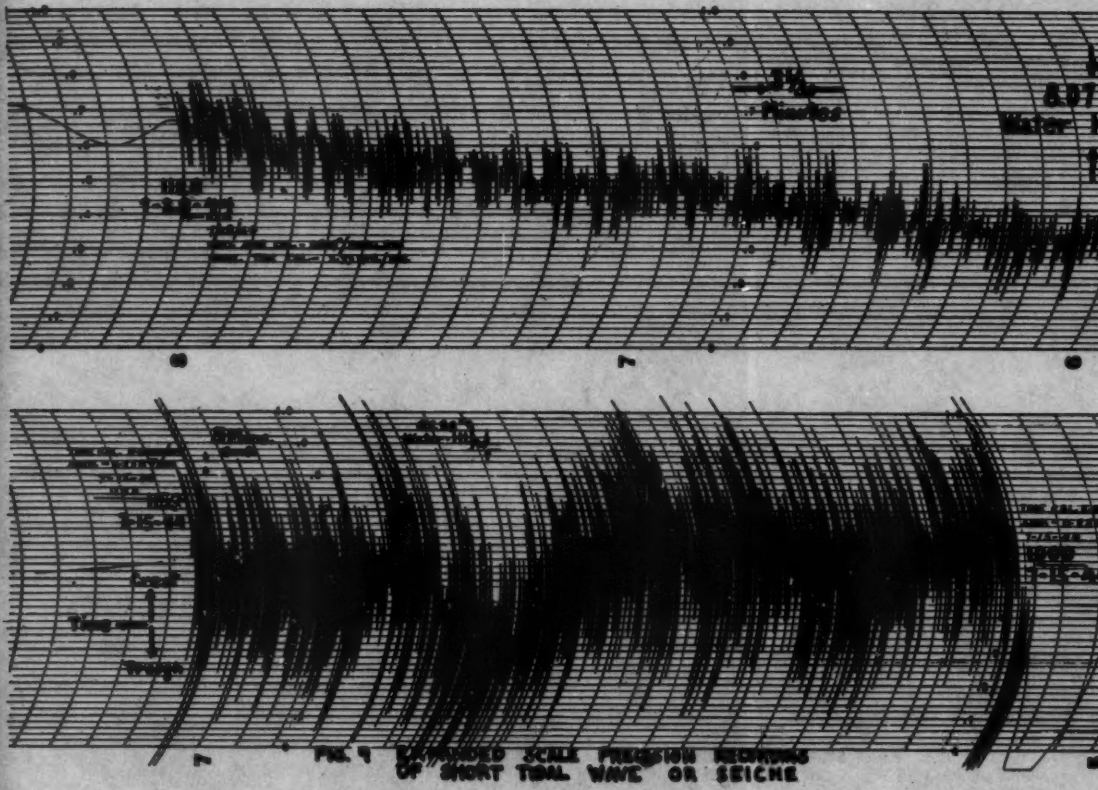
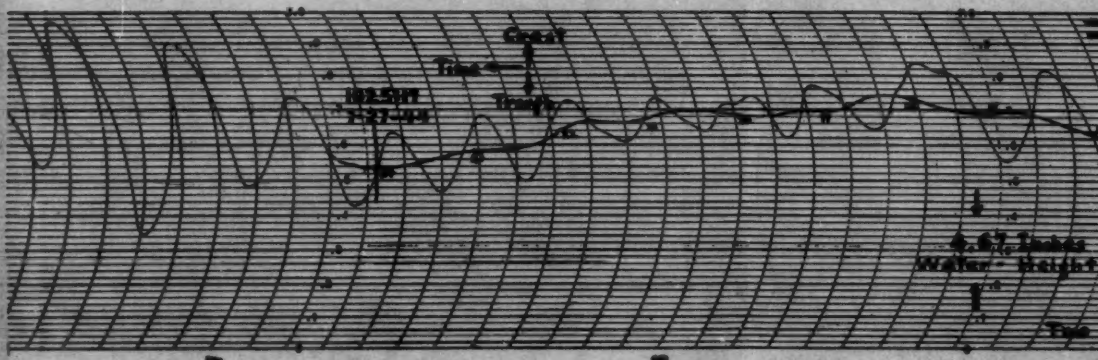
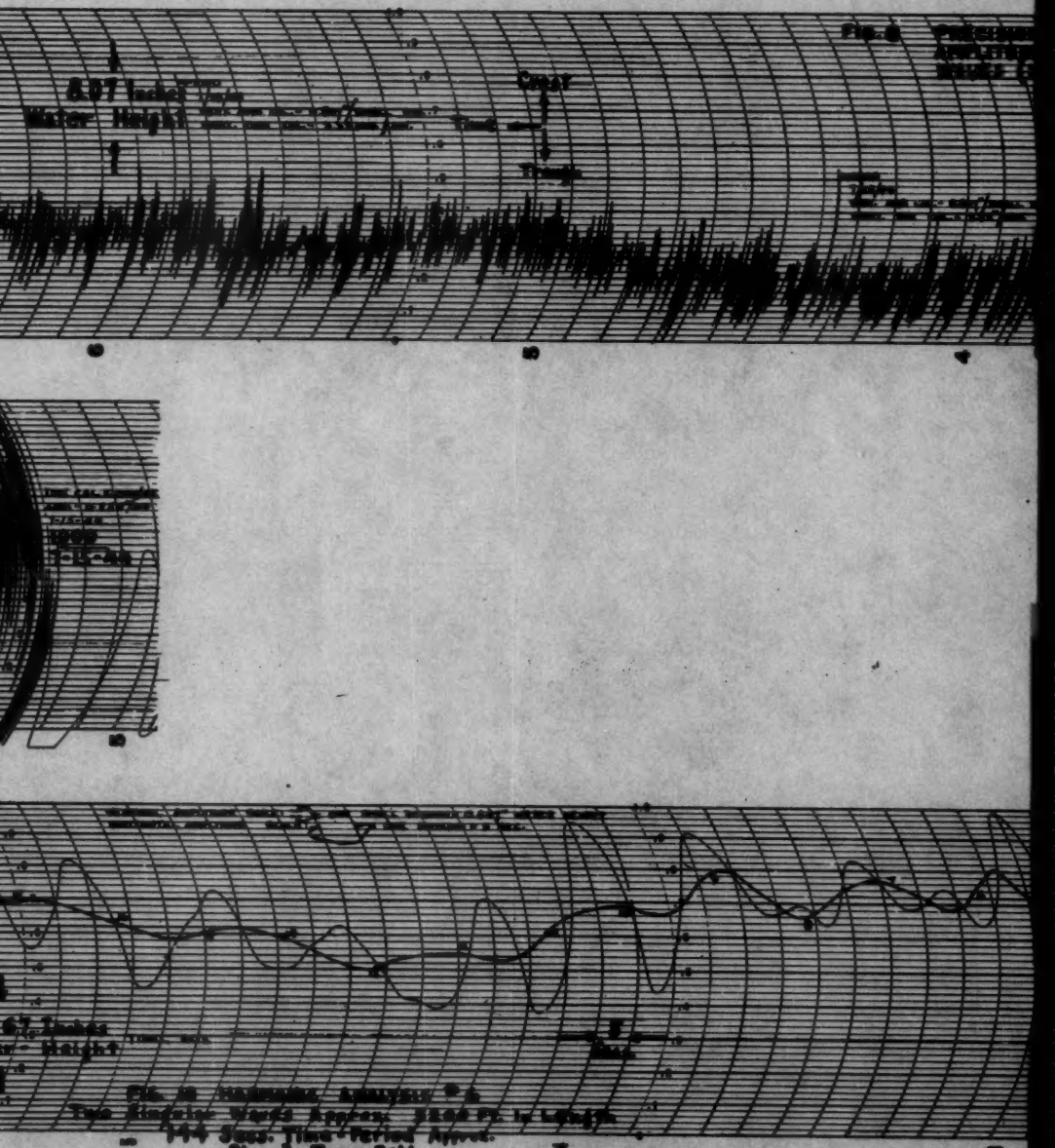


FIG. 9 ENLARGED SCALE TRACINGS OF SHORT TIDAL WAVE OR SEICHE





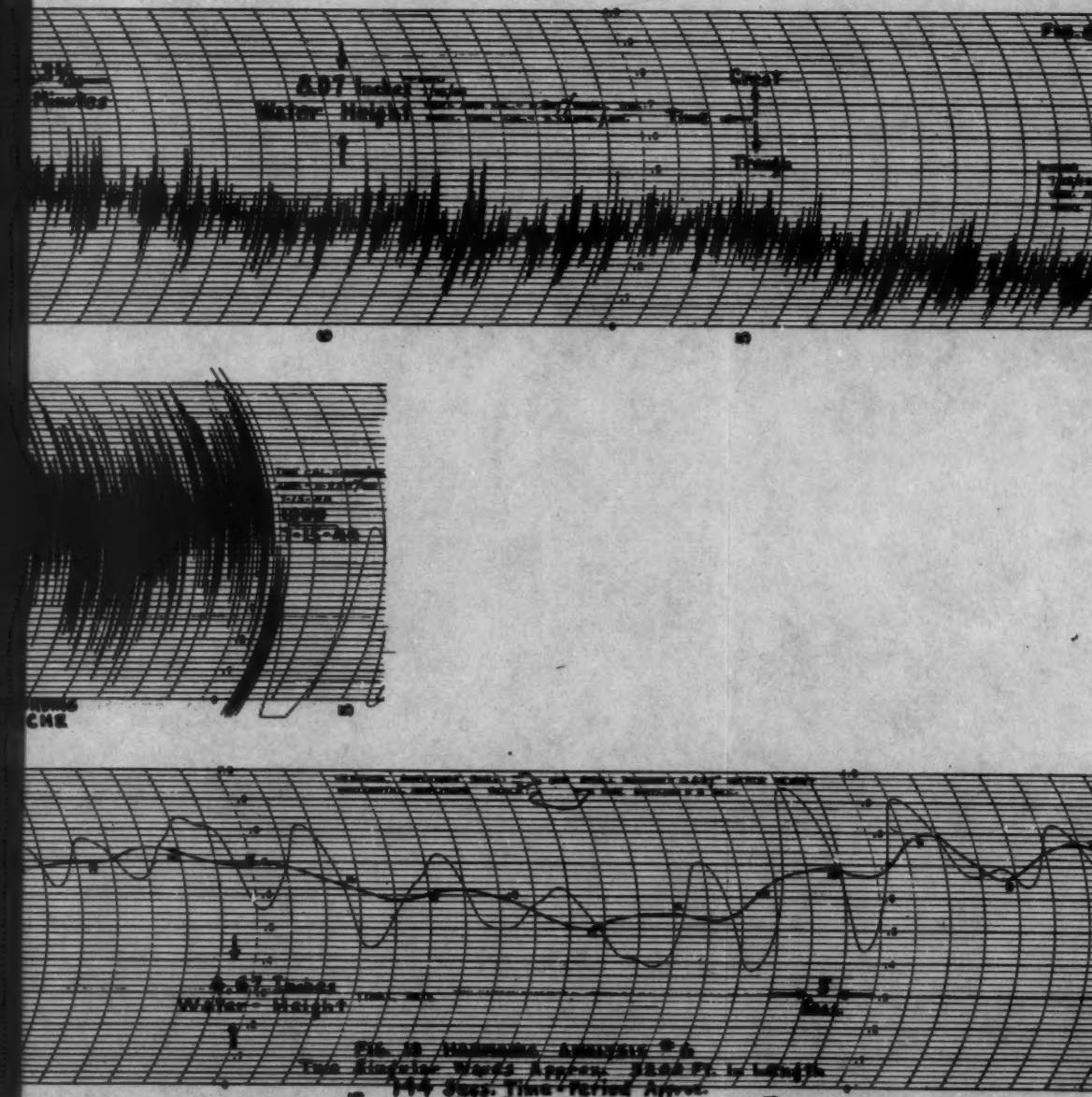
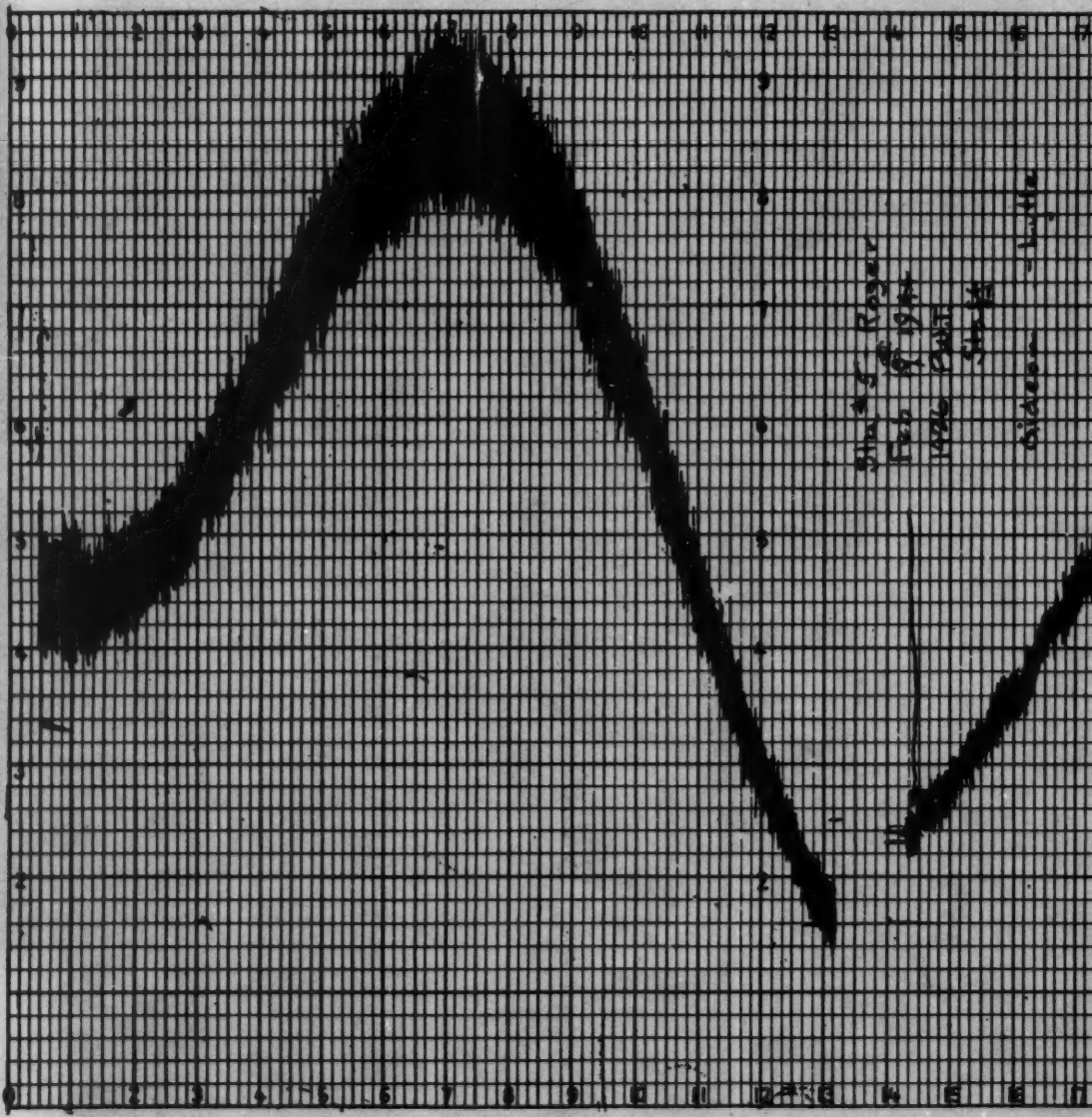
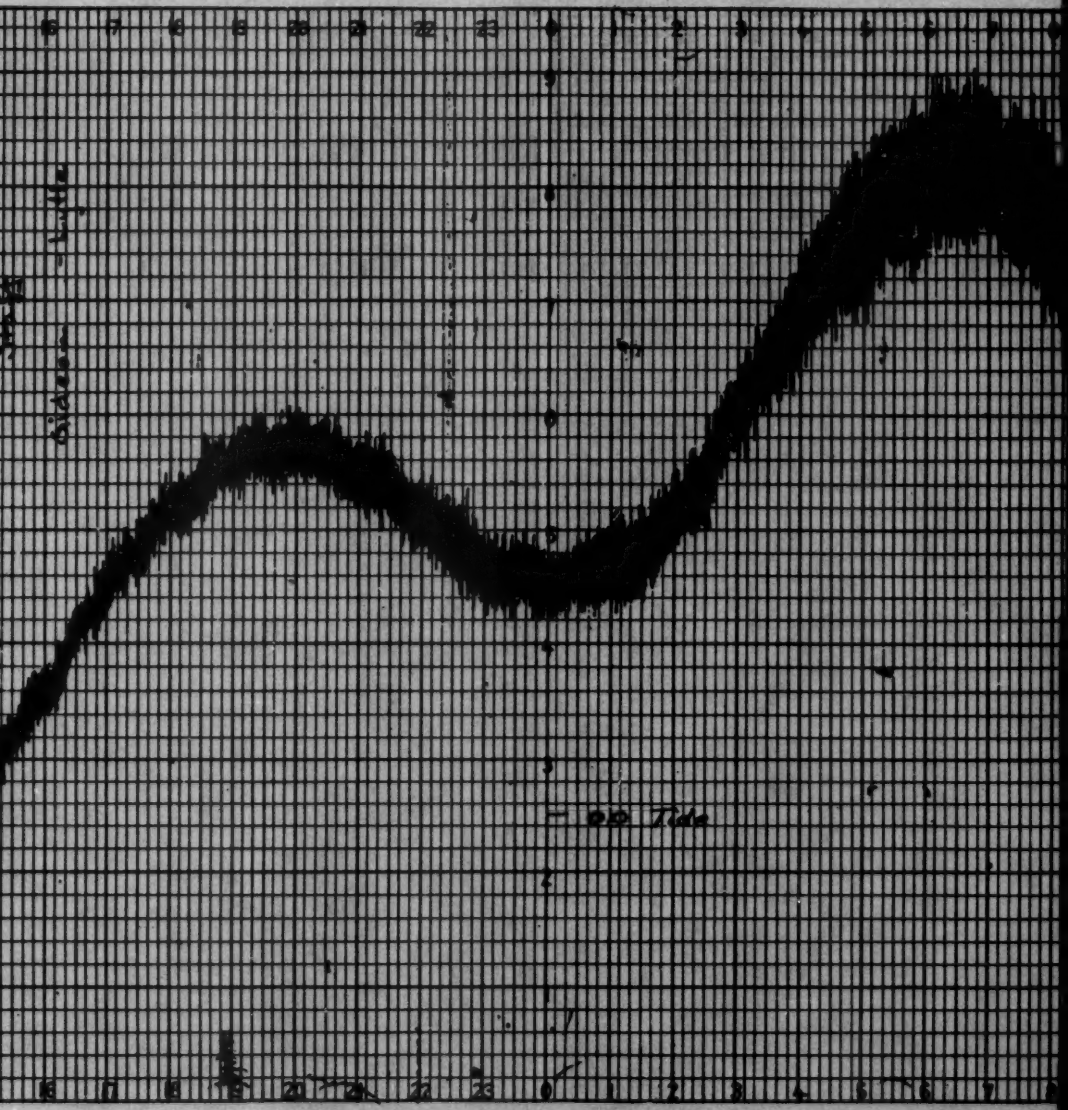


Fig. 18 Harbourne Analyser
Type Angular Wave Approx. 3600 ft. in Length
144 Secs. Time Period Approx.
Pier 3 - Term Isld.





TIME SCALE — 10 MINUTES PER SMALLEST D

FIGURE 12.—TYPICAL WATER-STAGE RECORDING MADE IN LOS ANGELES-LONG BEA

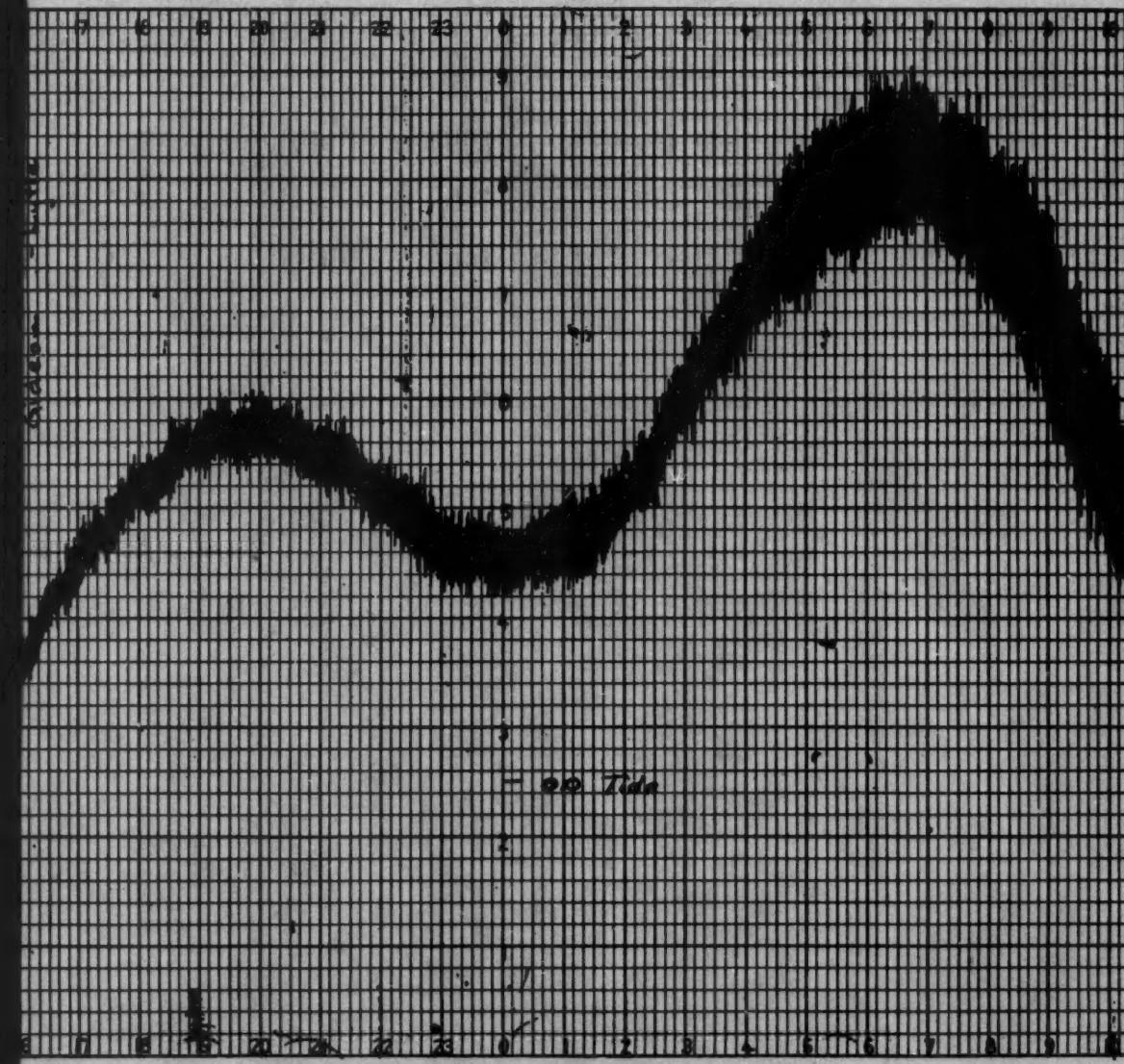
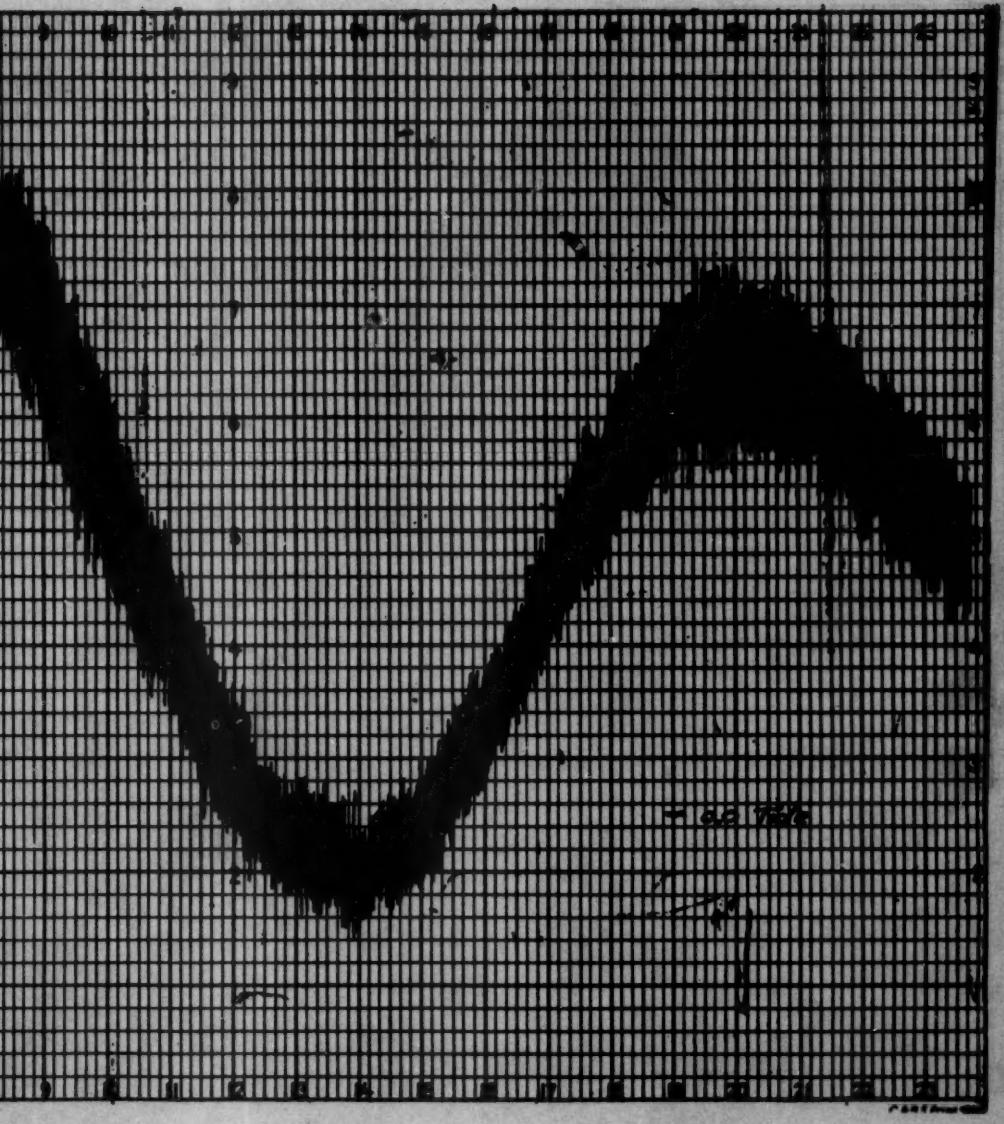


FIGURE 12.—TYPICAL WATER-STAGE RECORDING MADE IN LOS ANGELES-LONG BEACH HARBOR.



VISION

H HARBOR.

FIGURE 15.—AERIAL VIEW TOWARDS SANTA CATALINA ISLAND OF THE WEST SECTION OF LOS ANGELES-LONG BEACH HARBOR AND OUTER BREAKWATER, TAKEN FROM 12,000 FEET ELEVATION. OCEAN SURGE CLEARLY VISIBLE ENTERING HARBOR THROUGH WEST ENTRANCE.

FIGURE 16.—AERIAL VIEW OF CENTRAL SECTION OF OUTER BREAKWATER. NORMAL SURGE CLEARLY VISIBLE IN PACIFIC OCEAN AND ENTERING WEST HARBOR ENTRANCE (LOWER RIGHT).

FIGURE 17.—AERIAL VIEW OF EAST SECTION OF OUTER BREAKWATER. NORMAL SURGE CLEARLY VISIBLE IN PACIFIC OCEAN AND ENTERING HARBOR THROUGH EAST ENTRANCE IN RIGHT CENTRAL FOREGROUND.

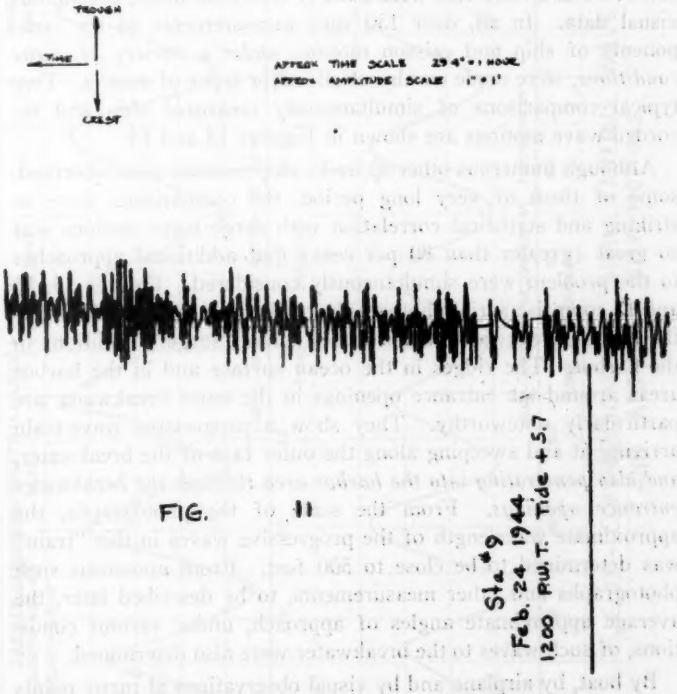


FIG. 11

FIGURE 11.—TYPICAL MARIGRAM FROM LATEST TYPE TIDE-GAUGE IN LOS ANGELES-LONG BEACH HARBOR (Photo Reduced 2¼ Times).

synchronized in time-period of their rolling and "charging" motions with the wave-motions being continuously recorded at points adjacent to the ships. A series of clocked measurements of fore-and-aft horizontal, athwartships horizontal, vertical and rolling motion of ships and drydock-caissons were made from immediately adjacent piers while simultaneous recordings of incident wave-motions were also made. A sufficient number of observers and recorders were used to obtain accurate, continuous visual data. In all, over 130 such measurements of the components of ship and caisson motion, *under a variety of surge conditions,* were made on almost all major types of vessels. Two typical comparisons of simultaneously measured ship and recorded-wave motions are shown in Figures 13 and 14.

Although numerous other sporadic ship-motions were observed, some of them of very long period, the comparisons were so striking and statistical correlation with surge wave-motions was so great (greater than 80 per cent) that additional approaches to the problem were simultaneously considered. Figures 15, 16 and 17 comprise aerial photographs of the area inside and outside the harbor breakwater, taken under active "surge" conditions in the harbor. The ridges in the ocean surface and in the harbor areas around the entrance openings in the outer breakwater are particularly noteworthy. They show a progressive wave-train arriving at and sweeping along the outer face of the breakwater, *and also penetrating into the harbor area through the breakwater entrance openings.* From the scale of the photographs, the approximate wavelength of the progressive waves in this "train" was determined to be close to 500 feet. From numerous such photographs and other measurements, to be described later, the average approximate angles of approach, under various conditions, of such waves to the breakwater were also determined.

By boat, by airplane and by visual observations at many points on the outer breakwater and over the harbor area, it became quite apparent that the approaching Pacific Ocean surge waves were arriving in the breakwater area with a definite velocity of propagation and wavelength. Further tests to confirm this by measurement were made. Figures 18 and 19 depict a method of calibrated boat runs against and with the direction of the progressive wave trains. Theoretically, by clocking the times for the boat to go from crest to crest in each direction, wave-velocity and wave-

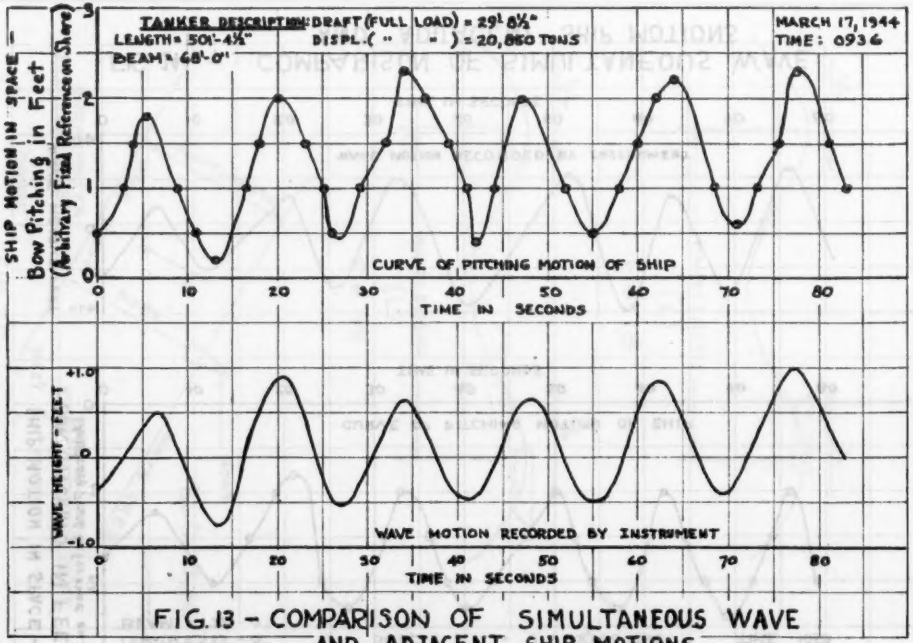

TANKER DESCRIPTION: DRAFT (FULL LOAD) = 29' 8½"
LENGTH = 501'-4½"
DISPL. (") = 20,860 TONS
BEAM = 68'-0"
MARCH 17, 1944
TIME: 0936
SHIP MOTION IN SPACE — Bow Pitching in Feet (Arbitrary Fixed Reference on Shore)
CURVE OF PITCHING MOTION OF SHIP
TIME IN SECONDS
WAVE HEIGHT - FEET
WAVE MOTION RECORDED BY INSTRUMENT
TIME IN SECONDS

FIG. 13 – COMPARISON OF SIMULTANEOUS WAVE AND ADJACENT SHIP MOTIONS

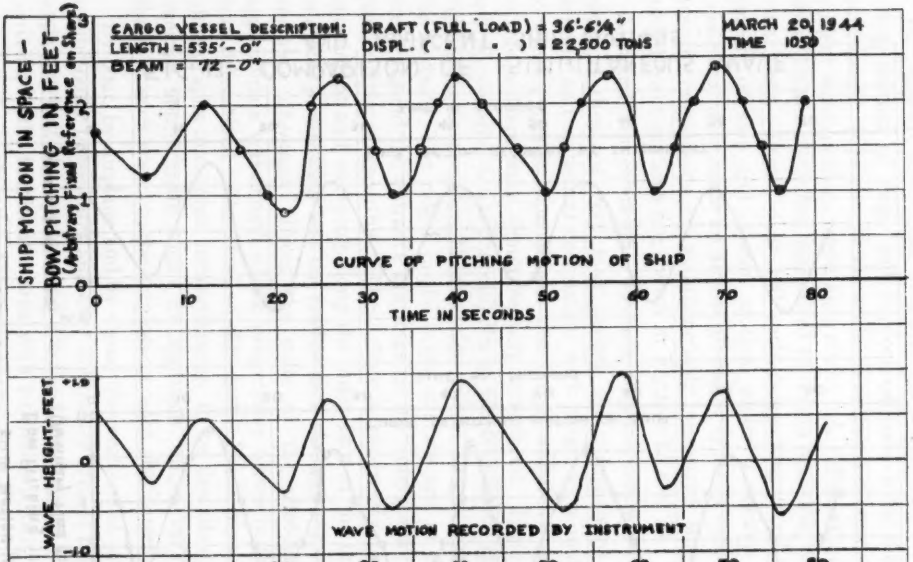

CARGO VESSEL DESCRIPTION:
LENGTH = 535'-0"
BEAM = 72'-0"
DRAFT (FULL LOAD) = 36'-6¼"
DISPL. (" ") = 22,500 TONS
MARCH 20, 1944
TIME 1050
SHIP MOTION IN SPACE - BOW PITCHING IN FEET (Arbitrary Fixed Reference on Ship)
+2
0
CURVE OF PITCHING MOTION OF SHIP
0 10 20 30 40 50 60 70 80
TIME IN SECONDS
WAVE HEIGHT - FEET
+10
0
-10
WAVE MOTION RECORDED BY INSTRUMENT
0 10 20 30 40 50 60 70 80
TIME IN SECONDS

FIG. 14 - COMPARISON OF SIMULTANEOUS WAVE AND ADJACENT SHIP MOTIONS

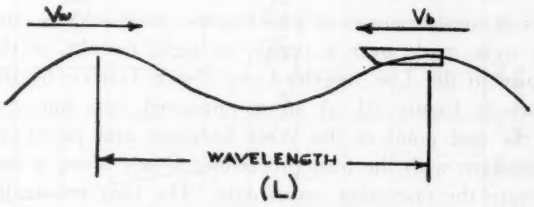


FIG. 18

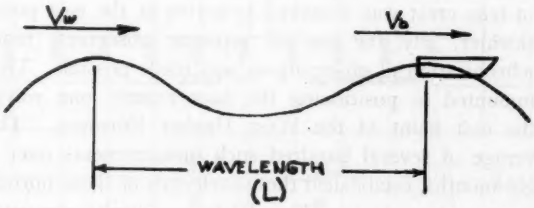


FIG. 19

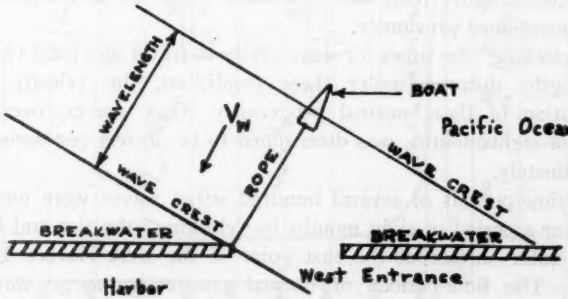


FIG. 20

length could be calculated, provided boat-velocity could be accurately known and maintained. Practically, and because of the sharp curvature in the region of interest (wavelength close to 500 feet) in the curves of Figure 21, this method was not at all satisfactory, and the results obtained with it were rejected. Better methods were of necessity devised.

Very accurate measurements of wavelengths, wave-velocity, and wave-period were made over a period of eight months at the West Entrance to the Los Angeles-Long Beach Harbor by the method shown in Figure 20. A stout, measured, rope-line was fastened to the east point of the West Entrance and payed-out over the boat-stern with the boat proceeding slowly along a line perpendicular to the oncoming wavefronts. The boat eventually achieved a position (maintained by suitable rudder and propeller action) such that a wave-crest passed under the boat at the same instant that a true crest was observed to arrive at the east point in the breakwater. By the use of separate observers, most accurate synchronization of observations was made possible. This procedure amounted to positioning the boat exactly one wavelength off the east point of the West Harbor Entrance. The statistical average of several hundred such measurements over a period of eight months, established the wavelength of these normal progressive surge waves to be 520 ± 10 feet. Similar measurements were made over two wave lengths, also. This value is a more precise figure than that obtainable from the aerial photographs mentioned previously.

By "clocking" the times for wave-crests to travel one (and two) wavelengths distance under these conditions, the velocity of propagation of these normal progressive surge waves, over a period of eight months, was determined to be 36 feet per second, approximately.

The time-periods of several hundred surge waves were measured over a period of eight months by "clocking" the rise and fall of the water surface at the east point of the west Harbor Entrance. The time-periods of normal progressive surge waves ranged, almost entirely, from 12.5 to 16.5 seconds, statistically averaging about a value of 14.5 seconds. *The time-periods so measured were corroborated by simultaneous instrument recordings.*

Still referring to Figure 20, it is obvious that the direction of

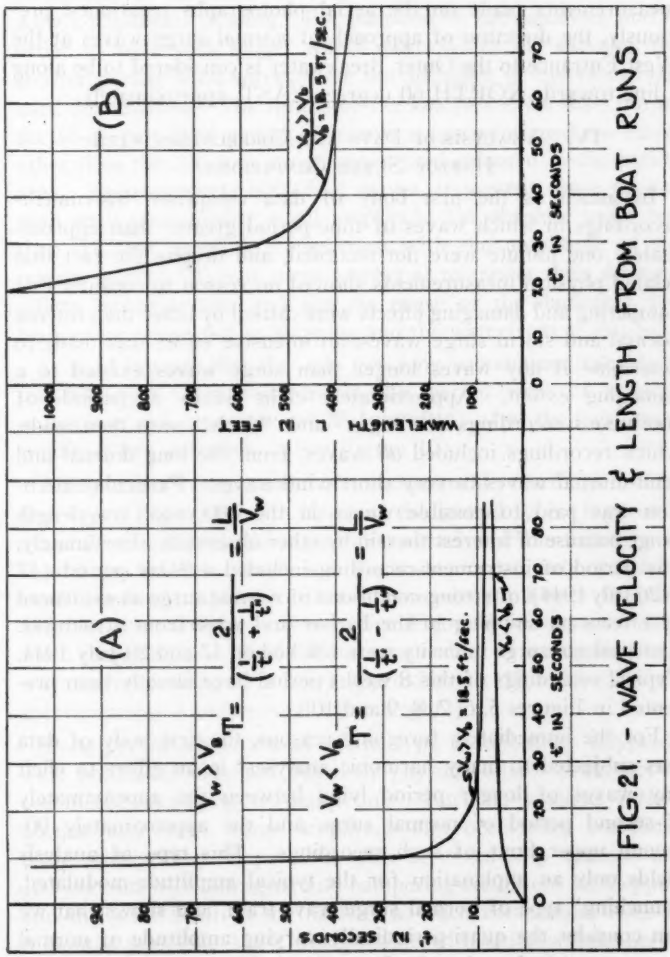


FIG. 21. - WAVE VELOCITY & LENGTH FROM BOAT RUNS

the extended rope-line gives the bearing of the line of approach of these Pacific Ocean surge waves. Based upon several hundred such measurements over an eight months period and upon measurements made on the aerial photographs mentioned previously, the direction of approach of *normal* surge waves at the West Entrance to the Outer Breakwater is considered to be along a line towards NORTH 60 degrees EAST, approximately.

IV. Analysis of Data and Correlation with Harbor Surge Conditions.

Inasmuch as the first body of data comprised instrument-recordings in which waves of time-period greater than approximately one minute were not recorded, and despite the fact that related physical measurements showed no reason to consider that hampering and damaging effects were caused by other than intense normal and storm surge waves, an intensive effort was made to determine if any waves longer than surge waves existed to a damaging extent. Approximately eight weeks of periods of continuous recordings at "Sugar" and "Uncle" were then made, which recordings included *all* waves, from the long diurnal and semi-diurnal waves to very short wind-waves. Particular attention was paid to possible waves in the 6000-foot wavelength range because of interest therein by other observers. Fortunately, this period of instrument-recording included a 9-day period (17 to 26 July 1944) of strong conditions of normal surge as evidenced by effects on shipping in the harbor and waterfront structures. Two peaks of surge intensity were reached on 17 and 20 July 1944. Typical recordings of this 8 weeks period have already been presented in Figures 5, 6, 7, 8, 9 and 10.

For the immediately foregoing reasons, the first body of data was subjected to many harmonic analyses[8] in an effort to elicit any waves of longer period lying between the approximately 15-second period of normal surge and the approximately 90-second upper limit of such recordings. This type of analysis yields only an explanation for the typical amplitude-modulated, "bunching" type of normal surge wave-train, and shows that we can consider the quasi-periodically varying amplitude of normal surge waves in Los Angeles-Long Beach Harbor to be caused by the continuous interference of three predominant harmonic wave-components whose time-periods are very close to each other and grouped about an average value of approximately 14.5 seconds.

Also, for the reasons already mentioned, harmonic analyses of of the empirical type (24-coordinate method of Runge[9] and Scarborough[10]) were conducted on numerous typical instrument-recordings of the precision type. These recordings contained all wave components existing. For purposes of such analysis, and in order to make such longer waves more evident, if existing, the shorter period wind-waves were not included in the recordings used for analysis. The results of the analyses show that, during the 17-26 July 1944 period of intense surge mentioned, no waves other than the 520-foot surge waves predominated in the harbor area. Approximately 6000-foot waves, if present, were present with such questionably small amplitude (in absolute value less than 0.2 feet, small in comparison with the amplitude of the intense normal 520-foot surge waves) as to render their dynamic effects inconsiderable and not the cause of the damaging and hampering action felt on shipping and the waterfront at this time.

It is of interest that the only two long waves unmistakably in the 6000-foot range recorded in this period were of singular occurrence and occurred (on 27 July 1944) after the local surge condition had subsided. (See Figure 10).

Many exact comparisons of measurement-recordings of the first type (Figure 3) with those of the second type (Figures 5, 6 and 7) established the engineering veracity of the first type of recordings in their range and the existence of only relatively small amplitude, sporadic waves of wavelength longer than the 520-foot normal surge waves. (See Figure 8 for a partial tide-wave recording and Figure 9 for a recording of a singular long-wave corresponding to a wave approximately 10 miles long and approximately 4 inches total wave-height!)

Long-period seiche waves of the type described by Leypoldt[1] were habitually observed but their physical effects could not be considered damaging or hampering. Further, whereas *intense* normal and storm surge amplitude and occurrence correlated more closely than 80 per cent with observed ship motions and 100 per cent with damaging and hampering conditions on shipping and the waterfront in the harbor, no such correlation, over an 8 months period in 1944, was possible with existent long-period seiches, or long-period progressive waves.

Rather, it appeared that seiche phenomena were superimposed, chimera-like and inconsequentially, upon the persistent, larger

amplitude pattern of the normal and storm surge waves. There is, however, no doubt as to the existence and excitation of the numerous seiche patterns of which Leypoldt[1] has written so admirably. It yet remains for someone of such broad experience in this field to correlate the arrivals of large amplitude normal surge waves in Los Angeles-Long Beach Harbor with the far distant meteorological disturbances which are probably the focus of their creation.

The correlation of the determined 520-foot wavelength, approximately 15-second time-period and approximately 36 feet per second wave-velocity of the normal surge waves, with the values calculated for this area by classical hydrodynamics, is most striking. Space does not permit a theoretical discussion of these and related calculations at this time.

Finally, the values of water-particle acceleration, velocity and displacement in normal surge waves were calculated from classical hydrodynamics.[11, 12] Space does not permit their inclusion here. Similar calculated data on 6000-foot progressive waves were prepared for purposes of comparison with those of the 520-foot surge waves. Energy analyses, not contained herein, and these data are believed to further demonstrate analytically that *intense* 520-foot surge waves because of their very great relative (and absolute) amplitude and synchronous time-period, are the cause of the damaging and hampering water-wave effects experienced frequently in Los Angeles-Long Beach Harbor.

V. Conclusions.

From the measured and recorded data obtained in the Los Angeles-Long Beach Harbor area for the first eight months of 1944, and the analyses thereof, it is considered that the following evidence has been gathered. Needless to say, eight months is too short a term in which to gather anything more than that portion of the fundamental data described. Further investigation would be expected to fill in the needed missing gaps and furnish corroboration or modification of present concepts. Considering the over-all benefits to the harbor and to future harbor work resulting therefrom, an extended study would seem to be justifiable and economical, especially in the post-war period. Figure 22 is an idealized picturization, insofar as possible, of the following conditions considered to obtain in this Harbor.

FIG 22 - IDEALIZED SURGE PATTERN
(M-T-S)
Pacific OCEAN
SAN PEDRO BAY
LONG BEACH LIGHTHOUSE
S.E. STORM SURGE
PIER #3
NORMAL SURGE
TERMINAL ISLAND
BREAKWATER
DIRECTION OF TRAVEL OF NORMAL SURGE WAVES
DIRECTION OF TRAVEL OF CENTER OF STORM SURGE
EAST ENTRANCE
WEST ENTRANCE
WATER OUT
2

1. The disturbing normal surge waves in this area are progressive waves, approximately 520 feet in wavelength, 14.5 seconds in average time-period, 36 feet per second in velocity of progress.

2. Because the disturbing surge waves in this harbor are progressive waves, and not "standing waves", because their time-period varies slowly and continuously, and because their time-period is not derived from the geometric configuration of the Los Angeles-Long Beach Harbor, the local *disturbing* surge is definitely not "seiche" and should not be called by that name.

3. Probably originally created by meteorogical disturbances at far distant points at sea, surge waves arrive at the Outer Breakwater along a line of travel towards approximately NORTH 60 degrees EAST normally.

4. The average maximum wave-height of the normal surge waves in the ocean just outside the Outer Breakwater, varies from 6 to 8 inches during a "quiet" period to 6 to 8 feet during an "active" period of disturbing normal surge.

5. The wave-heights of successive normal surge waves vary between maximum and minimum values every 2 to 4 minutes; in other words, the waves appear in large amplitude groups with alternating small amplitude groups. Such a pattern presents the appearance of "bunching" or "amplitude-modulation" common to interfering wave-trains of very nearly equal wavelengths.

6. The Outer Breakwater itself effects an amplitude reduction of 80 to 90 per cent "from outside in" of those ocean surge waves incident upon the outer face of the breakwater and permeating through it into the Harbor area. In other words, the Los Angeles-Long Beach breakwater permeability to normal ocean surge waves is only between 10 and 20 per cent. There is some evidence that the western section of the breakwater is even less permeable to normal surge.

7. The normal ocean surge waves enter the Los Angeles-Long Beach Harbor through the navigational openings in the Outer Breakwater.

8. Normally, the ocean surge waves enter the Harbor area through the West Entrance (navigational opening) in the Outer Breakwater, to a lesser extent through the East Entrance, and to an inconsiderable degree through the open east end of the Harbor.

9. Variation to the northward in the angle of approach to the

Outer Breakwater of the ocean surge waves progressively increases the contribution of the East Entrance to the disturbing surge effects experienced in the Harbor.

10. The West and East Entrances in the Outer Breakwater, through which the disturbing progressive ocean surge waves enter the Harbor, seem to act as partial point-sources of somewhat circular wave-fronts which radiate over the Harbor area. The progressive circular wave-patterns which emanate landward from the two Entrances overlap and interfere with each other in the Harbor.

11. The West and East Entrances also seem to behave as partial planewave-front directional radiators in that maximum amplitudes in the circular wave-patterns inside the Harbor area occur on those portions of the wave-front arcs whose radii of curvature are parallel to and approximately co-linear with the direction of travel of the ocean surge waves arriving at the Entrance openings outside the breakwater. Under normal surge conditions this line extended landward from the center of the West Entrance proceeds generally towards the eastern portion of Terminal Island, a well-known surge "problem area".

12. The average ratio between average surge amplitude just outside the Outer Breakwater and at the eastern portion of Terminal Island is roughly 1.5:1.

13. The direction of approach of ocean surge waves varies moderately under normal conditions, moving from a line of approach towards NORTH 60 degrees EAST to more nearly due NORTH, the more intense the local southeast and south winds become. In times of severe local southeast storms, the local surge pattern changes radically. The angle of approach at the breakwater changes to a direction towards NORTH 20 degrees WEST, approximately. The time-period of storm surge reduces to a range of 6 to 9 seconds; the wavelength decreases proportionally (to almost one-half the 520-foot wavelength of normal surge waves); the wave-height increases to a maximum which depends upon the intensity and duration of the storm, how long the local storm has been in progress, and upon the direction of the storm winds.

14. In times of recent severe storms in this area, southeast storm surge waves have reached amplitudes of 10 to 12 feet in the ocean just outside the breakwater, and amplitudes of 3 to 5

feet on Terminal Island. Because of their shorter wavelength, the intense southeast storm surge waves apparently attenuate more rapidly than do the normal surge waves in their progress across the Harbor.

15. In times of intense southeast storm surge, due to the changed angle of approach to the breakwater from the ocean, storm surge waves "drive through" the East Entrance, predominantly, in a line almost directly towards the eastern portion of Terminal Island. Consequently, in times of intense normal surge and storm surge, the eastern portion of Terminal Island is definitely a problem area.

16. The protection afforded by such promontories as Long Beach Pier A and the Watchorn Promontory, San Pedro, makes the waters behind them relatively surge-free except in times of most intense normal or storm surge.

17. In so sheltered a space as Bethlehem Slip, Terminal Island, and well up the San Pedro Main Channel, the wake of ships passing causes local disturbances which normally exceed the effects of surge waves in these areas. However, in times of severe southeast storm surge, because of the alignment of the San Pedro Main Channel entrance with respect to the West Entrance, the storm surge "drives through" the West Entrance and proceeds, with high amplitude, in a direct line towards this normally surge-free channel, enters it and produces disturbing effects therein.

18. Such bottom contours as the beach shoaling off the city of Long Beach and elsewhere in the Harbor, do much to absorb, retard or deflect such portions of the surge waves as pass over them in their progress across the Harbor area. This factor, as well as the angle of approach of the surge waves, is considered, for example, to be the main reason for the very insignificant contribution made by the open east end of the Harbor to damaging surge effects in the Harbor area.

19. The separate circular attenuation, damping, directional and reflection coefficients of this Harbor have not yet been determined, nor was this achieved in this investigation. Hence, it is only possible to present the foregoing over-all data at this time.

20. In over 130 timed measurements, the components of motion of many types of ships in the Harbor area correlated better than

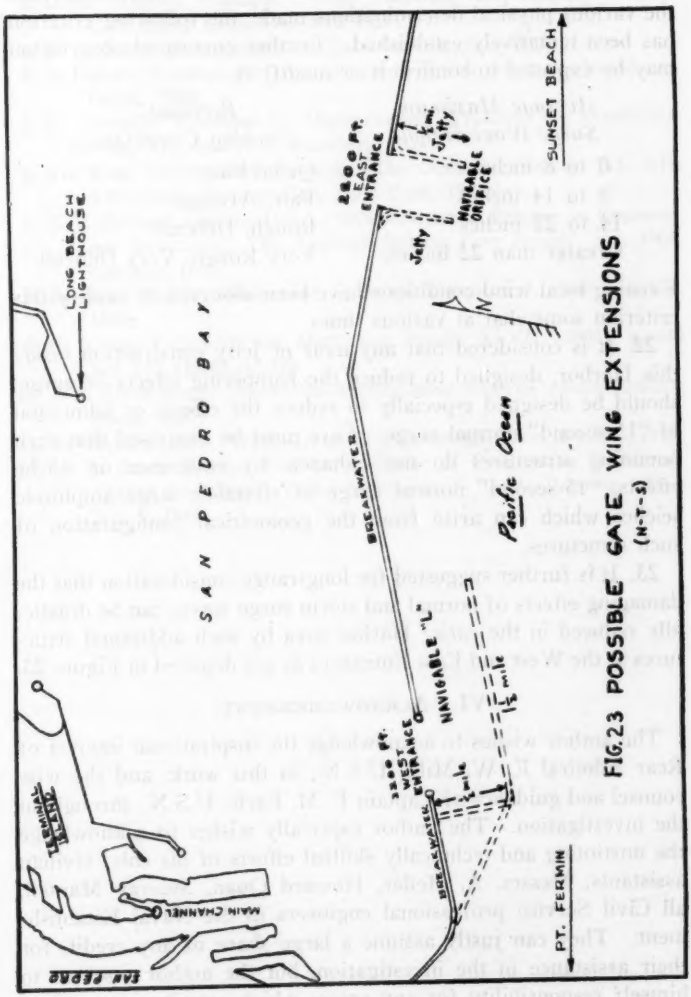


FIG. 23 POSSIBLE GATE WING EXTENSIONS (N-T-S)

80 per cent with the measured vertical and calculated horizontal water-particle motions of the progressive surge waves.

21. On the basis of 8 months of measurement-recordings and the various physical determinations made, the following criterion has been tentatively established. Further correlated observation may be expected to confirm it or modify it.

Average Maximum Surge Wave-Height	*Berthing, Docking Conditions*
0 to 8 inches	Quiet, Easy
8 to 14 inches	Fair, Average
14 to 22 inches	Rough, Difficult
Greater than 22 inches.	Very Rough, Very Difficult

Existing local wind conditions have been observed to modify this criterion somewhat at various times.

22. It is considered that any mole or jetty construction *inside* this Harbor, designed to reduce the hampering effects of surge, should be designed especially to reduce the effects or admission of "15-second" normal surge. Care must be exercised that such bounding structures do not enhance, by resonance or seiche effects, "15-second" normal surge or stimulate large amplitude seiches which can arise from the geometrical configuration of such structures.

23. It is further suggested for long-range consideration that the damaging effects of normal and storm surge waves can be drastically reduced in the *entire* Harbor area by such additional structures at the West and East Entrances as are depicted in Figure 23.

VI. Acknowledgement.

The author wishes to acknowledge the inspirational interest of Rear Admiral E. W. Mills, U.S.N., in this work, and the wise counsel and guidance of Captain F. M. Earle, U.S.N., throughout the investigation. The author especially wishes to acknowledge the unstinting and technically skillful efforts of his chief civilian assistants, Messrs. A. Heller, Howard Quan, Murray Mantell, all Civil Service professional engineers in the Naval Establishment. They can justly assume a large share of any credit, for their assistance in the investigation, but the author reserves to himself responsibility for any errors which may have crept into this resume. Credit is due Mr. F. F. Mead, Civil Service professional engineer, for the excellent aerial photographs.

VII. Bibliography.

1. Harry Leypoldt, "California Seiches and Phillipine Typhoons", United States Naval Institute Proceedings, June, 1937;
2. D. E. Hughes, Principal Civilian Assistant, Los Angeles District, U. S. Army Engineers, circa 1924;
3. F. A. Forel, Investigations on Lake Geneva, circa 1875;
4. A. Defant, "Dynamische Ozeanographie", Einfuhrung in der Geophysik, Berlin, 1929;
5. H. Thorade, "Probleme der Wasserwellen", Probleme der kosmischen Physik, XIII and XIV, Hamburg, 1931;
6. Sverdrup, Johnson and Fleming, "The Oceans", Chapters XI, XII, XIII, XIV, Prentice-Hall, New York, 1942;
7. D. D. Gaillard, Captain, U.S.A., "Wave Action in Relation to Engineering Structures," The Engineer School, Fort Belvoir, Va., 1904, reprinted 1935;
8. Byerley, "Fourier Series and Spherical Harmonics", Ginn and Co., 1893;
9. C. Runge, Zeitschrift für Math. and Physik, XLVII (1903), p. 443 and LII (1905), p. 117.
10. J. B. Scarborough, "Numerical Mathematical Analysis", John Hopkins Press, 1930;
11. Sir Horace Lamb, "Hydrodynamics", Cambridge Press, 6th Ed., 1932;
12. U. S. Beach Erosion Board, "A Study of Progressive Oscillatory Waves in Water," Tech. Series, U. S. Govt. Printing Office, No. 1, 1941.

DISCUSSION OF ARTICLE BY CAPTAIN J. E. HAMILTON, U.S.N., ON "POST-WAR RESEARCH AND THE NAVY."

By Lieut. Commander Donald L. Herr, U.S.N.R.*

The subject of Captain Hamilton's recent article is of salient importance because the policy adopted by the Navy regarding its post-war research activities will in large measure determine our ability to cope with any possible future emergency.

The article deals with some of the more fundamental aspects of the Naval research organization and treats them at a fairly high level in the administrative structure. It is desired, in the following discussion, to emphasize two aspects of the problem which, based on the present writer's experience in educational, industrial and Naval research, appear significant. This discussion has been stimulated by Captain Hamilton's article and is presented now while the large number of people presently engaged in such work are still available for comment and consideration. It is hoped that discussion such as this will aid in precluding undue curtailment of Naval research and development in the post-war period.

Assuming that the high-level policy matters settle themselves somewhat along the lines that Captain Hamilton suggests, there still remain the two basic problems of providing and maintaining suitable research personnel in Naval research organizations and providing and maintaining a vigorous, progressive Naval research administration, at all administrative levels, in the post-war period.

We shall take up the first item, namely, that of continually providing and maintaining an adequate number of especially qualified and competent research engineers who reflect in their education, experience and thinking the modern status and present trends of the field in which they are expert. These people will no doubt be entirely or almost entirely individuals working as civilian employees of the Navy Department. The government, hence, will be in competition with private industry in securing the

* Office of Research & Inventions, Navy Department, Washington, D. C.

services of these men from the standpoints of salary offered, rate of progress and ultimate professional attainment possible. While it is true that there are those broad altruistic motives which provide incentives for creative Naval research work to many men in time of war and to fewer men in times of peace, it is not believed that military expediency can rely in any country upon the patriotic impulses of relatively few individuals in the national group to provide the definite technical and technological output required in peace-time. On the other hand, the men associated in sufficient number with the Naval organization in its research establishments must remain abreast of and preferably ahead of word-wide technical advance, especially in the field of military applications. It would therefore seem to be a basic matter of policy of national security that the government, through whatever acts of Congress may be necessary to insure same, would provide sufficient incentives in the form of adequate salaries, opportunities of advancement and high professional status to men in such service so that the government will continuously attract, against the competition provided by industry, at least a portion of each year's crop of superior research and development engineering graduates. The existing advantages of such government service are already well known but the point remains that added incentives, over and above those of security, ultimate pensions and stability must be provided in order that all the progressive elements of creative, applied research may be actively maintained and even further stimulated in the Naval field. These extra advantages of participation in government research, in the Navy, for example, must be assured of active publicity in the colleges throughout the country so that each year's output of engineering graduates may seriously consider service in government research as professionally attractive a life-time opportunity as any provided by industry.

On the other hand, there are certain incentives for service which civilian employment in Naval research can provide which apparently are not too commonly found to exist in similar employment in industry. For example, the liberal rights attendant upon an individual inventor in employment in Naval research are a striking exception to the general practice in industry. Further, from the professional standpoint, the solution of engineering research and development problems pertaining to shipboard application of new developments in peace-time affords a field of activity not generally available to the industrial researcher

as his life work. The importance of this sort of work and of the individual's share in this work are considerably accentuated in war as well as in peace. Hence, Naval engineering research and development could be considered to possess all of the attributes which an industrial professional career of very high standard possesses.

We now turn to the second of the two problems which it is the purpose of this paper to discuss: that of continually providing and maintaining progressive, dynamic administration of Naval research and development at all levels of administration, in the post-war period. It is presumed that Naval research and development will continue to flourish on a much more expanded scale and will embrace a much wider field of activity than existed in the peace period between World Wars I and II. One of the lessons of this war is considered to be that there is no field of research and development which does not find its place and application somewhere, even to a modified degree, in the Naval or military technological program.

In this province, we can draw heavily upon the experience and practices of modern industry. An examination of the lists of the directors and administrators of research in the organizations of most industrial corporations, large and small, shows that the men in these positions, in the great majority, have themselves been at some previous time successful researchers in their own right. At least insofar as the research and development aspect of Naval duty is concerned, it seems fairly evident that the administrators of such research cannot attempt to be masters of all technical and operational aspects of the Navy. The need for individuals familiar with all general projects of the Navy and having the broadest possible view will probably be restricted to only one or two such persons and these few only for administration of top policy matters.

Again, as in the field of industrial research, it is believed that in these top directors continuing to be highly qualified technical officers who have progressed up through the channels of active Naval research, it will thereby be insured that these individuals themselves will continue to adequately represent and be familiar with the "know-how" and "wherewithal" of the current status and needs of the profession. It is believed that from this level and only by this means, may the goals be set and the research trends interpreted on not only a Naval but a national and world basis as

well. Progress in science and in technology apparently recognize no measured boundaries and any delimiting of their horizons would only serve to insure tragedy from the beginning.

In order that all links of the chain may be of equal strength the foregoing remarks can be made to apply, in somewhat modified form, as the requirements for Naval administrators of Naval research at inferior levels. In each case a Naval research administrator should arrive at that position after having once been a successful researcher himself and after having experienced a period of appreciation of the operational problems involved. This, it seems, will insure his being truly expert in one field with the opportunity, by the very additional qualities which he has demonstrated as an administrator, to then become familiar with and appreciate more fully the related problems in other fields of Naval technology. Under these circumstances in industrial research, economy, direction and progress have been found to result and to be imparted to organizations in which these potentialities for creative growth are made to exist.

Research men who are not administrators or who do not wish to become administrators will be quite evident in their leanings. Potential administrators in research will be just as evident in their research abilities and in giving evidence of their administrative talents. Their own output of applied, successfully executed research and the morale and following of the group of which they form a part will attest to their qualities of leadership. Natural assumption of research administrative duties by such men should follow. The ultimate good of the organization as a whole will be reflected in and by the growing responsibilities of such men.

It is believed that, in a Naval research and development establishment in which the personnel and administration are sponsored and nurtured under these conditions, much will have been accomplished in maintaining in the post-war period that type of progressive Naval research which will stand this country well in any future time of need. A Naval research organization having these ingredients can insure and materially aid in the rapid synchronization of the industrial research effort with the Naval requirements of any such future time.

The foregoing are only one individual's concepts regarding two facets of the basic problem. Perhaps these comments may serve to evoke further discussion by those vitally interested in the problem of post-war research and the Navy.

NOTES.

INDEX TO NOTES.

CURRENT PRACTICE IN MARINE GEARCUTTING.
—Transactions, Institution of Engineers and Shipbuilders in Scotland, March, 1945.

LIGHT ALLOYS FOR MARINE ENGINES.
—Transactions, Institute of Marine Engines, March, 1945.

NEW WARSHIP CONSTRUCTION.
—Journal of the Royal United Service Institution, November, 1944.

CURRENT PRACTICE IN MARINE GEARCUTTING.

This paper, by A. W. Davis, of the Fairfield Shipbuilding and Engineering Co., Ltd., has been slightly condensed and reprinted from the March, 1945 issue of the Transactions of the Institution of Engineers and Shipbuilders in Scotland.

As in most other active spheres of engineering at the present time, the fast tempo of war and the requirements of the general situation have combined to produce a concentration of thought and application in the practice of marine gearcutting, the aim of which is the design and manufacture of a product of superior quality. In this country the marine gear has shown no very conspicuous advance in design during the last twenty years, and the progress that has been made in the technique of production is to a great extent limited. Nevertheless, having regard to the high degree of immunity from serious trouble and the general reliability for which the gears produced during this period have been renowned, it is permissible to query in what respects any appreciable improvement may be achieved.

The general tendency towards higher steam pressures in the quest for lower fuel consumption has, in effect, made desirable and often necessitated an increase in turbine speed, thereby calling for the reintroduction of the double-reduction gear so emphatically rejected by most shipowners in this country twenty years ago. For reasons that will be referred to later, this type of gear calls for greater precision of manufacture—in fact, a precision that was not generally attainable at the time of its unpopularity—and this very need constitutes one of the requirements for progress. There are two further respects in which advancement may be made, namely, in the reduction of gear noise and in the production of more compact and lighter units in relation to the power transmitted. The latter requirement is of particular significance in naval vessels, in which powers tend ever to increase while the demands on space become more severe.

The general lines on which progress is being made can be divided under the following headings:

1. Gear layout design.
2. Gear tooth design.
3. Precision in gearcutting.
4. Improvement of tooth surfaces by post-hobbing processes.
5. Gear measurement.

Tooth surface finish and accuracy of contact are linked with design in that, subject to certain qualifications, the greater the precision achieved the greater the tooth loading for which the gears may be designed. Such progress cannot be made with confidence until it can be assured that all gears to be manufactured to a particular design will comply with a specific minimum degree of precision. The lack of any such defined standard of comparison up to the present time has largely stultified the adoption of heavier tooth loading in taking advantage of greater manufacturing precision, and gears that have been finished to a high standard have, in consequence, carried a margin of safety that might well be considered excessive.

Accepting the desirability for a defined and rather stringent minimum standard of accuracy to permit of progress in design and quieter operation, consideration must be given to the competing claims for two standards as opposed to only one, the suggestion being that two standards of differing quality might ensure a more economically stable situation, having regard to the following facts:

(a) That provision and maintenance of super-accurate hobbing machines is costly, and that, generally speaking, a gear cut on less accurate machines should be obtainable at less cost to the purchaser.

(b) That gears to be cut on the lower grade machines would be designed on a correspondingly lower basis of tooth loading, so that they should be equally acceptable to the classification societies.

The argument against this conception may be stated

(c) That when first-class machines are available, all interests concerned would be desirous of obtaining the best possible product.

(d) That if the first-class machines are standing idle, competitive prices would probably show little, if any difference between the two qualities.

(e) Any difference that might exist would be to some extent minimized by the larger proportions that would be entailed by the more conservative design basis for the lower quality gears.

These three points weigh the scale heavily in favor of the adoption of one standard only, but it must be recognized that the latest requirements for naval gears are extremely severe, and, with the inevitable forthcoming reduction in naval construction, it may be found uneconomical to make more than a few machines acceptable by this standard, thus tending to amplify argument (a) and to make argument (d) less generally applicable.

It is now the intention to refer in more detail to each of the five avenues of progress already indicated, keeping to the broad lines of the trend of modern practice in gearing manufacture rather than dwelling on details of construction and design.

Gear Layout Design.

So far as single-reduction gears are concerned, little comment is called for. The principal distinction in type is defined by the fitting, or omission, of center bearings on the pinions. The difficulty of maintaining uniform wear is a disadvantage associated with the center bearing, and its accommodation involves a much heavier construction of the main wheel. To avoid the feature wherever possible, designs with two bearing pinions have been developed to the utmost, and it seems probable in many instances that the optimum ratio of face width to pinion diameter for this type of gear has been exceeded, the bending and torsional distortion of the pinion at full power actually producing localized loading in excess of that which would have obtained with a pinion of narrower face width. It is generally felt now that the ratio of face width (including gap) to pinion pitch circle diameter should not exceed about 2.5.

In the realm of the double-reduction gear many different designs have made their appearance at various times and in various countries, and it is only proposed to mention four representative types.

First is the type most commonly employed in this country and known here as the "Interleaved" gear (Figure 1). One of its disadvantages is weight, the main wheel and gearcase being of massive construction; another, and what can be a more serious disadvantage is the lack of flexibility between the primary and secondary gears, the effect of irregularities in the pitch of one being transmitted and superimposed upon the effects of the other. Such effects, moreover, may be of considerable magnitude, associated with tooth loading rising possibly to several times the driving torque load with the indifferent gearcutting that was common at the time when this type of gear reached the depths of its unpopularity. Even with the best modern standard of cutting, the aforementioned inherent characteristics of the gear cause the superimposed pitch error loading to rise to quite a substantial fraction of the full power load, and recogni-

tion must be made of this fact in the design stage. A variation of this type of gear is common in America and is shown in Figure 2, both types being known over there as "Nested." Since this second type is of American origin it is, however, proposed to associate it alone with the American title, and in so doing the recent recommendation of an Admiralty committee is being adopted. It will be seen that the nested gear differs from the interleaved type in that the primary elements straddle the secondary, instead of the secondary straddling the primary. The advantage of the nested design over the interleaved is weight reduction, due

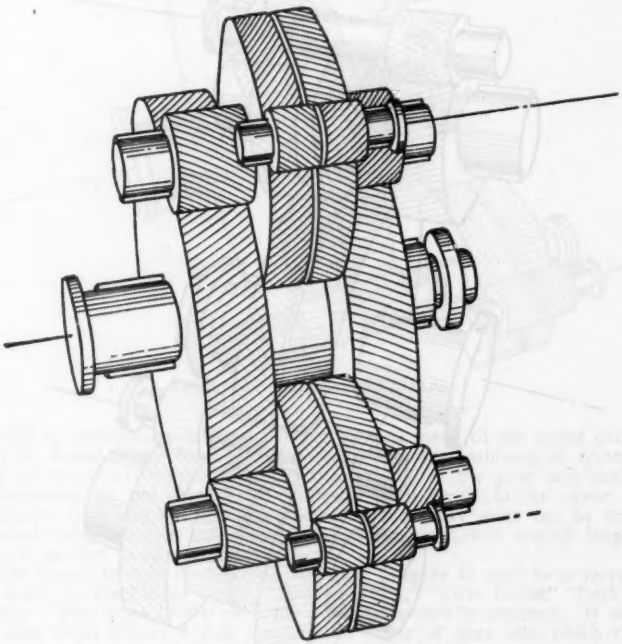


FIGURE 1.—INTERLEAVED GEAR.

both to the more compact construction of the main wheel and to the consequent reduction in the proportions of the gear case, the housing of the primary elements being somewhat overhung from the main structure. For naval work, this consideration is sufficient to direct favor to the nested type, but it will be appreciated that the gear suffers from the same defects of inertia and lack of flexibility as the interleaved design, and, moreover, a certain problem is introduced in the provision of bearings for the long primary pinions having widely separated helices; the latest method is to provide the pinion with three bearings, the pinion shaft between each helix and the center bearing tapering from the diameter at the root of the teeth to the journal diameter. Some companies have specialized in this class of gear in naval and merchant construction, but

among some other American firms the design has its severe critics, some of whom suggest that the maintenance of alignment is difficult. This would appear to depend upon the constructional rigidity of the gearcase in any particular design.

A third type of gear, unusual in marine practice in this country, is the Articulated design, illustrated in Figure 3, and commonly employed by one American firm in merchant construction. It will be seen that the primary and secondary units are separated, the drive between them being

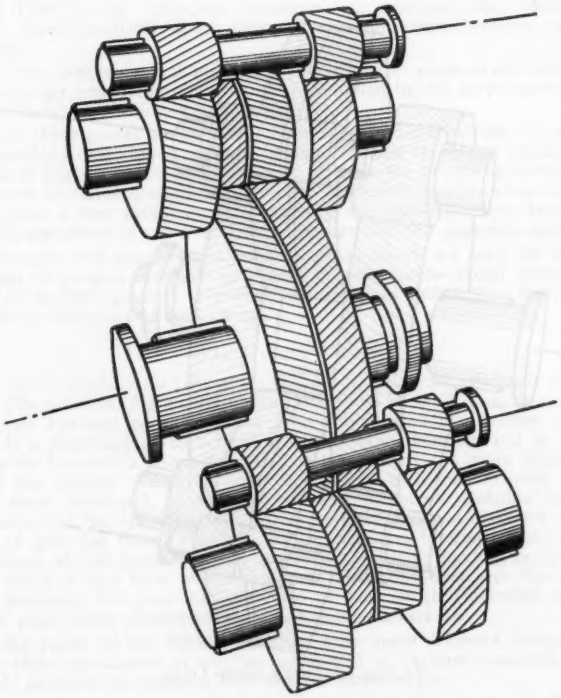


FIGURE 2.—NESTED GEAR.

taken by a quill shaft arranged to provide torsional flexibility and to permit of a certain degree of malalignment between the primary and secondary gears. It will be noted that the purpose of the quill drive is not to counter any torsional oscillation effects, such as was the object of quill shaft "nodal drives" to primary pinions, as introduced in this country some twenty years ago. Extensive analysis in the United States has shown that, with double-reduction gear systems, theoretical torsional oscillation amplitudes exceed the actual recorded amplitudes by many times, due, it is now thought, to damping in the bearings, and the system of torsional balance, which formed the foundation of the "nodal drive," is now re-

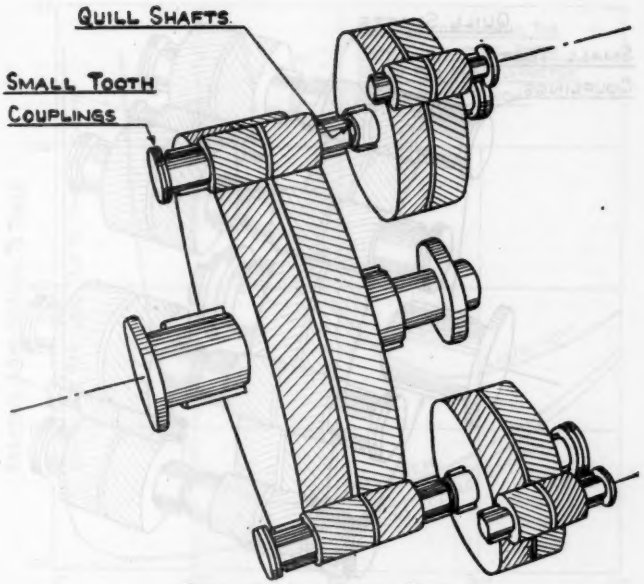


FIGURE 3.—ARTICULATED GEAR.

garded as unnecessary but, nevertheless, a real merit of the nodal drive may be found in the flexibility it introduces in the turbine-gear system, this to an extent compensating for the rigidity of the gear unit itself.

Returning to the Articulated gear, a claim made in its favor is simplicity in production since primary and secondary units can be progressed independently, but considerations of weight and overall length rule it out for naval construction.

The fourth type of double-reduction gear (Figure 4) goes by a variety of names in the United States, "Divided Train," "Twin Drive," "Locked Train." The latter title is probably the most generally accepted. It will be seen from Figure 4 that the parallel trains of gear into which the drive from each primary pinion divides are locked, in that they finally mesh into the same secondary gearwheel, but as the drive from the primary to the secondary elements is carried through quill shafts, adequate flexibility is provided to accommodate, within reasonable limits, the differing cumulative pitch errors in the parallel trains without greatly disturbing the equal balance of power transmitted by them. The reduction in face width, made possible by this duplication of the drive, has a considerable influence in the reduction of space and weight, the latter being to some extent offset by the greater number of component parts. Since a single unit taking the drive from two turbines comprises 5 wheels, 6 pinions, 4 quill-shafts, 12 pairs of flexible coupling elements and 22 bearings, it follows that the man-hour expenditure on gears of this design is considerable and that they are costly to produce. Difficulties in production lie in obtaining an equal division of power between the gear trains in parallel and in maintaining equal tooth thicknesses throughout those

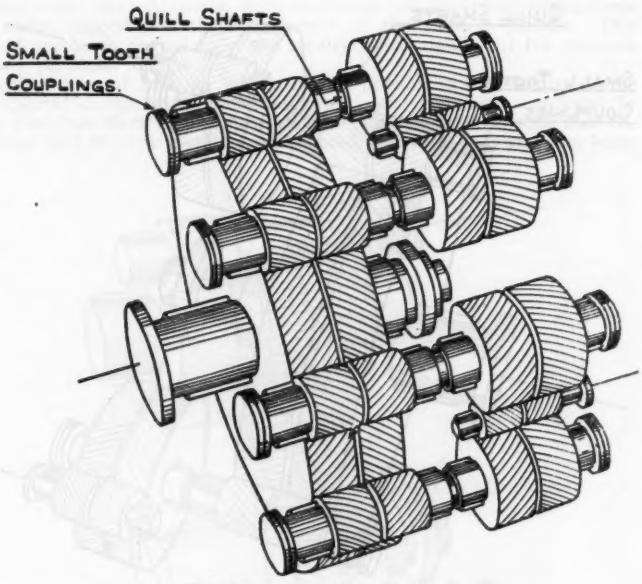


FIGURE 4.—LOCKED TRAIN GEAR.

trains to ensure that the balance as achieved in the ahead direction is preserved in the astern direction of rotation. The balance is usually accomplished by providing the secondary pinions, primary wheels and quill shaft couplings with numbers of teeth each containing a high prime factor which is not common; then by disconnecting a quill shaft coupling the angular position of a primary wheel may be adjusted with respect to a secondary pinion by a desired amount in such a way that the coupling can be remeshed in a new register of the teeth; for this purpose the flexible couplings concerned with the adjustment must be finished to an extremely high degree of pitch accuracy. Apart from these admittedly real difficulties, the design accommodates itself very well to gear case construction, accessibility is good and the general effect is pleasing. The gear is widely used in American naval construction today.

GEAR TOOTH DESIGN.

This is such a very wide subject that it is impossible to deal with it adequately in a few paragraphs, but it would be appropriate to summarize the principal views that are prevalent today without becoming involved in any of the mathematics with which the problem abounds.

Helical Angle. Practice in this country has, up to the present time, stabilized on 30 degrees. In the United States angles up to 45 degrees have been used quite extensively, partly with a view to reduction of gear noise, although opinion is divided as to its merit in this connection. On the other hand, some American interleaved designs show an 18 degree angle to reduce axial loading on the main wheel structure. The effective tooth loading varies with respect to helical angle in the manner indicated

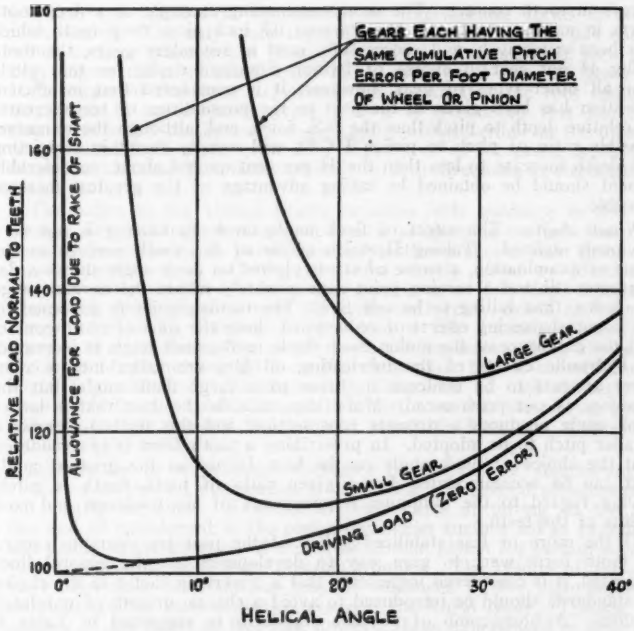


FIGURE 5.—EFFECT OF HELICAL ANGLE ON TOOTH LOADING.

in Figure 5, from which it will be seen that for any given standard of gear cutting the optimum value of the helical angle varies for different gear designs between quite wide limits. It is, therefore, suggested as unreasonable that the helical angle should be stabilized for all gears. An angle of 15 degree is possibly the very minimum that should be adopted, having regard to the necessity for maintaining a number of teeth in simultaneous contact.

Tooth Depth and Pitch. For a constant value of the ratio of the meshing depth to the normal pitch of the teeth, the gross effective length of tooth in instantaneous contact remains virtually constant. If the ratio can be increased, the total contact length is increased and the specific loading on the tooth surface is reduced for a given total load and face width. The factors which limit such an increase are partly dynamic and partly geometrical, the former being the high ratio of sliding to rolling which may be introduced at the extremity of the contact zone by a deep tooth form, and the latter being considerations of minimum desirable root radius, minimum desirable tip thickness of teeth and avoidance of undercutting. All the factors are helped by a low ratio of pitch to the P.C.D. of the pinion, and, consequently, from the point of view of tooth loading, the smaller the pitch the better. Such reduction is, however, limited in its turn by root strength. A secondary benefit, resulting from an increase in the depth ratio, is increased flexibility of the tooth, which is fundamentally important in the absorption both of the effects of pinion deflection under bending and torsional load and of minor manufacturing

errors in tooth contact. The most outstanding example of a deep tooth form in marine practice is the Parsons 0.7 inch pitch deep tooth which has been extensively and successfully used in secondary gears, the tooth being 44 per cent in excess of British Standard depth for that pitch. For all other types of gear, however, it is considered that insufficient attention has been given in the past to the possibilities of teeth greater in relative depth to pitch than the B.S. form, and, although the minimum feasible ratio of pitch to pinion P.C.D. will usually result in restricting the depth increase to less than the 44 per cent quoted above, considerable benefit should be obtained by taking advantage of the greatest increase possible.

Flank Angle. The effect of flank angle on tooth loading is not very obviously defined. Taking Hertzian stress of the tooth surface as the basis of examination, a curve of stress plotted on flank angle shows a flat characteristic and a turning point approximately within the normal range of choice, thus failing to be selective. The turning point is governed by the counterbalancing effects of reduction in both the zone of tooth contact and the curvature of the pinion tooth flank, as the flank angle is increased. If hydraulic effects of the lubricating oil film are taken into account there appears to be evidence in favor of a large flank angle, but the tendency is not pronounced. More important is the fact that a larger flank angle produces a stronger root section, and this tends to enable a smaller pitch to be adopted. In prescribing a tooth form it is considered that the choice of flank angle can be best defined as the greatest angle that can be accommodated for a given ratio of tooth depth to pitch, having regard to the minimum requirements of tip thickness and root radius of the teeth.

If the more or less stabilized policy of the past ten years in regard to tooth form were to give way to development on the general lines indicated, it is considered imperative that a governing factor in the choice of standards should be introduced to avoid a chaotic growth of unrelated profiles. A honeycomb of possible standards is suggested in Table I,

TABLE I. PROPOSED HONEYCOMB OF TOOTH STANDARDS.

Relative meshing depth taking B.S. of (0.636 × normal pitch) as 1	Flank Angle	Axial pitch of hob				
		0.40 in.	0.50 in.	0.60 in.	0.70 in.	0.80 in.
1.0	22½°	E	O	X	O	E
1.1	20°	O	O	O	O	O
1.2	18°	O	O	O	O	O
1.3	16½°	O	O	O	O	O
1.4	15°	O	O	O	Y	O

O = proposed standards *for adoption as essentially required.*

E = existing standards.

X = tooth form closely approximated by existing standard having relative meshing depth of 1, ⅞ inch pitch and 22½ degrees flank angle.

Y = tooth form closely approximated by existing standard having relative meshing depth of 1.44, 0.70 inch pitch and 14½ degrees flank angle.

based on the general statements made above. It must be emphasized that it is not suggested that all the tooth forms shown should be adopted, but that the table should be regarded as a guide in the provision of any new standard. It will be observed that several of the extreme forms suggested are already closely represented by standards in current use,

but that the broad field represented by intermediate forms is wholly undeveloped. The only tooth form in current marine use in this country which finds no place in the "honeycomb" is the Parsons 7/12 inch pitch, 14½ degree flank, standard depth tooth, established some 25 years ago. It is suggested that later experience has shown that the root radius of this tooth is unnecessarily great, and it is considered that the space thereby occupied can be more economically utilized either in the direction of increased flank angle (vide 22½ degree) or of increased depth (vide 44 per cent increase) or intermediately in both directions as circumstances direct.

The policy in the United States provides little guidance in this connection, for, although development is now progressing on lines that could possibly be regarded as parallel to that outlined above, most firms have been conservative in adhering to forms closely allied or identical to Parsons' early standards. It is also surprising to find that, despite the degree of standardization over there, no uniformity exists in the tooth forms adopted by different firms. Also a fair amount of confusion appears to exist there, due to the tooth pitches having been allotted on the basis of diametrical pitch on the assumption that a constant helical angle is maintained. Subsequent modifications to helical angle are responsible for the confusion, and it is fortunate that the marine gearing industry in this country has adopted standards based on simple axial pitches of the hob.

Distribution of Addendum. Guidance on the factors determining the choice of addendum distribution is conflicting and several phenomena are not clearly explained. Evidence in many cases of pitting on the pitch line and of punishment to the pinion dedendum surface led to experiment and finally to the adoption of the all-addendum pinion for certain services, thereby obviating both these possible troubles. The results of this gear have been most creditable, as may be evidenced by the high degree of reliability of the main reduction gears of the Royal Navy. Trouble with scuffing of the tooth surface has, however, been experienced in some new gears (before "high" spots have worn off) in which the ratio of the tooth depth to the P.C.D. of the pinion has been relatively high, suggesting that the ratio sliding/rolling is tending to become excessive at the end of contact in recess. Should deeper teeth be employed this aspect would appear, in theory, to become more prominent, and consequently it is considered that a modified proportion of pinion addendum should be adopted between 50 per cent and 100 per cent of the meshing depth, preferably about 70-75 per cent. This would avoid high sliding rates at the end of the recess and also the relatively severe conditions which are believed to occur at the very commencement of approach in a 50 per cent addendum gear. Moreover, the proposal provides for a sufficient reduction in pinion dedendum to avoid undercutting, which is a factor to be considered when adopting a high ratio of tooth depth to pinion P.C.D. The phenomenon of pitch line pitting may be met with in all gears other than those of all-addendum form, but the extremely localized nature of the complaint seems to render it of relatively small significance, and in the United States, where the all-addendum form has never been adopted, the defect is accepted as apparently unavoidable.

Precision in Gearcutting.

The principal hobbing machine errors and the corresponding defects which they introduce in the work gear are shown in Table II, to which the following comments relate:

Pitch Error in Lead Screw. Many lead screws commercially finished are quite unsuitable for the stringent requirements of gear hobbing.

TABLE II. HOBBING ERRORS AND THEIR CAUSES

Source of Inaccuracy in Hobbing Machine	Nature of Principal Errors Produced				
	Defective Helical Angle	Undulations	Thinning of Teeth	Circumferential Pitch Inaccuracy	Defective Tooth Profile
Lead screw pitch error	C S				
Lead screw thrust abutment error or cyclic error in drive thereto		C S			
Main driving worm thread or worm wheel tooth profile error, thrust abutment error in worm or cyclic error in drive thereto		C S*	C		
Tooth to tooth error in the contact of creep ring and master wheel		C*			
Pitch error in master wheel (which is worm wheel in case of solid table machine)				C S	
Pitch error in creep ring relative to worm wheel teeth		C	C		
Periodic errors in gear train between hob and table or chuck		C S	C S		
Lack of uniformity in hob teeth, eccentricity of hob or its arbor, or cyclic error in hob drive			C S		C S
Lack of parallelism between axis of work and travel of hob carriage	C S				

* Undulations having crests in a series of lines parallel to the instantaneous line of contact, thus meshing in unison and being productive of a gear note. In the case of spur teeth these become wholly profile errors.

C shows error produced on Creep drive machines.

S. " " " Solid table "

Error in Thrust Abutment of Lead Screw, or Cyclic Error in Drive thereto. Extreme care in the machining and fitting of the thrust components and facings can ensure that errors due to this cause are kept within fine limits. Likewise the worm wheel driving the lead screw requires close attention.

Error in Thread of Main Driving Worm, in Profile of Worm Wheel Tooth, in Thrust Abutment of Worm or Cyclic Error in Drive thereto. Before commenting on this point the need is felt for emphasizing the basic characteristic of a creep machine. There are many requirements that have to be met in assigning the number of teeth to the worm wheel, creep ring and master wheel, but it is fundamental that the numbers should be all different and that they should contain no high common factor or conditions of worm phase similarity with table position will be repetitive.

The effect of a worm error in the drive of a solid table machine is to create undulations on the tooth surface parallel to the instantaneous line of contact and which therefore mesh in unison, the effect in the case of a spur tooth being one of profile error, and in the case of a helical tooth undulations running diagonally from tip to root and at an angle specified by the above condition. Consequently, even the smallest worm error will be the source of cyclic fluctuation in the drive with the very probable accompaniment of a shrieking note. In a properly designed creep machine the error is distributed so that the wave crests are not disposed in lines corresponding to the above description, thereby obviating cyclic fluctuation in the drive and minimizing noise. The effect of this distribution, plus the errors associated with the creep ring itself, is to produce a certain unevenness in the tooth surface, which limits to an extent the loading for which the teeth are suitable and which must be responsible for a considerable amount of gear noise of a generally non-cyclic character. It is on these grounds that the Americans have taken exception to the creep drive and reverted to solid table machines.

The success of the design of a creep machine is closely associated with the value chosen for what may be termed the "creep fraction." This quantity is defined as the fractional portion of the value wm/c where c is the number of teeth in the creep ring, m the number of teeth in the master wheel and w the number of teeth in the wormwheel. The creep fraction is distinct from the more commonly discussed but less informative creep ratio, $(c-m)/m$. The whole value wm/c represents the total number of turns of the worm during one complete revolution of the master wheel, and the fractional part of this value clearly represents the fractional portion of a revolution through which the worm turns for successive complete revolutions of the master wheel. For the sake of simplicity regard for the moment the case of the cutting of a spur gear. If the creep fraction is zero the machine is virtually non-creep and the particular instance already cited will be found to correspond with this condition, each successive revolution of the table corresponding with the same rotary position of the worm. If, on the other hand, the value approximates to zero, say 0.01, successive revolutions of the table will correspond with a slowly advancing movement of the worm, and 50 complete revolutions of the table will find the worm in a position 180 degrees from its starting position. Thus, with a feed of, say, 1/12.5 inch, an axial wave length of 8 inches would be produced on the gear tooth. Reference to a subsequent section dealing with post-hobbing processes will show that the presence of a long wave length is extremely undesirable unless lapping is introduced. Consequently, a value of the creep fraction approximately to zero is to be avoided.

It can also be shown that a value of the creep fraction approximating

to 0.5 has the same feature to condemn it as in the case of values approximating to zero, the effect being of lesser degree.

To a decreasingly important extent creep fraction of 0.33 and 0.67 and values approximating thereto should likewise be avoided, but it seems probable that the effect under discussion is reduced to insignificance for successive orders of creep fractions 0.25 and 0.75, etc. It is thus apparent that the choice of creep gears should be by no means haphazard and, in fact, it will be found that when attention is given to the above considerations and to the necessity of avoiding common factors in the numbers of teeth, it becomes a matter of some difficulty to achieve a desirable arrangement. It is furthermore necessary to avoid high prime numbers, since in the correction of the gear ratio in the drive to the worm a wheel embodying the same number of teeth would clearly have to be included.

These remarks have all been made with reference to a spur wheel, and they remain fundamentally correct for the cutting of a helical wheel, since the instantaneous contact of pinion to wheel occurs along a series of lines running diagonally from tip to root of the teeth within the axial band of the contact zone, each of which lines is generated by the hob at the same instantaneous position of the master wheel. A slight modification to the theory becomes necessary in regarding the *appearance* of undulations along the length of the tooth of a helical gear, and this applies also to the measurement of such undulations at constant tooth depth.

For such considerations the ratio wm/c must be multiplied by the factor $(1+f \tan \theta)/\pi D$ where f is the axial feed per revolution, θ is the helical angle and D is the pitch circle diameter of the gear being cut; the significant figure is the fractional portion of this multiple, representing as it does the fractional portion of the rotation of the worm between successive feeds in the cutting of two points at the same depth on any individual tooth. For the sake of reference this fractional portion will be referred to as the "helical creep fraction" and it will be observed that although the creep fraction is a constant for the machine the helical creep fraction varies with the diameter of the gear, its helical angle and the feed employed. On a solid table machine the creep fraction is zero, but the helical creep fraction will have a definite value, and correspondingly although the fundamental error of a solid table cut gear is a profile error at right angles to the instantaneous line of contact, giving no axial wave on a spur tooth, the apparent error along the length of a helical tooth is wave form. It may be added that the helical creep fraction exceeds or falls short of the creep fraction by an amount varying normally between about 0.02 for wheels to 0.10 for pinions.

Tooth to Tooth Error in the Contact of Creep Ring and Master Wheel. An illusion tends to be created by the contact of master wheel and creep ring teeth in that it would appear that the length of the zone of contact between an internal and an external gear is considerably greater than is actually so. The geometric construction involving the oblique line of contact holds good, however, for an internal gear and, bearing this in mind, it is seen that the zone of contact between the two elements is, in fact, quite short. Thus any error in the profile of the teeth will tend to produce a cyclic error of contact closely analogous in its effects to the worm error in a solid table machine. It is therefore absolutely essential that the radial depth of contact between master wheel and creep ring teeth should be broad so as to ensure uniformity of motion. This is sometimes difficult to achieve by reason of the fact that to mesh with an involute external tooth the internal tooth must have a hollow profile, and as this cannot be hobbed it must be produced with a form cutter. An alternative scheme is to hob the external teeth with a hob having suitably

curved flanks so as to give broad meshing with straight-sided internal teeth. This permits of hand correction being made to the pitch of these teeth as necessary, a performance which is impracticable with hollow profiles.

Pitch Error in the Master Wheel (the Worm Wheel in a Solid Table Machine). Comments relating to this point have been covered by previous remarks.

Pitch Error in Creep Ring relative to Worm Wheel Teeth. It is essential that pitch errors in the creep ring spur teeth should be regarded not on an absolute basis but in relation to the corresponding positioning of the worm wheel teeth. Errors in the creep ring teeth may be measured through the medium of the worm drive with a divided plate on the worm shaft and any necessary corrections may be made by hand work. One firm of hobbing machine manufacturers has recently perfected a system of production whereby the internal teeth of the creep ring are hobbed with a fly cutter, the creep ring being rotated by the worm and the normal driving shafts leading to it. To obtain the greatest benefit of this process the necessity has been stressed of arranging for the spur or bevel wheels which correct for the number of teeth in the creep gear to be arranged in the train of gears to the hob rather than in the train between the motor and the table. By this means it can be arranged that the shafts leading to the worm perform an exact number of revolutions for each revolution of the worm, thereby ensuring that the method of producing the creep ring teeth as described corrects not only for worm errors but also for cumulative and eccentricity errors in the gears leading to the worm driving shaft.

Periodic Errors in Gear Train between Hob and Table or Chuck. Comment has just been made on one important aspect of errors in the gear train, but it might be added that many intermediate wheels which in the past were often regarded as above suspicion by reason of the fact that they were "fast running" are in fact no longer immune now that such fine limits of precision are being sought in the final product. Mr. Couling has already drawn attention to the errors that tend to be associated with differential gears, and he has gone so far as to advocate their elimination. The Author does not share in this drastic condemnation of so useful a component, but it is certainly the case that they demand the closest attention to ensure adequate accuracy. Although the motion between the differential component and the main driving worm may be relatively fast running it is wrong to imagine that this action suppresses the effects of a slowly rotating differential worm wheel, which may have a considerable cyclic error producing a relatively long wave undulation on the teeth of the work gear. A serious aspect of errors of this nature is that they may combine with creep distributed worm errors to give axial bands of high spots comparable in their effect to the worm errors produced by a non-creep machine.

Lack of Uniformity in Hob Teeth, Eccentricity of the Hob or its Arbor, or Cyclic Error in Hob Drive. The methods of hob production and N.P.L. inspection are so good in this country that little need be said with regard to the accuracy of the hob teeth beyond noting that the closest attention is necessary in this connection to ensure correct profiling of the gear teeth. An aspect, however, that does call for more attention is the need for ensuring concentric running of the hob in the gearcutting machine. It is altogether out of proportion that elaborate precautions should be taken to check hob tooth form and spacing to limits of fractional thousandths of an inch and then that the hob should be permitted to run eccentric with a total throw of, say 0.003 inch. (The spacing of the hob teeth is of secondary significance and is of importance only by reason of the fact that the teeth are backed off and a disproportionately large space

between cutting edges is reflected by a comparatively small hob tooth. But for this effect the spacing would have no significance within reasonable limits of variation and this fact is frequently misunderstood.)

It has been suggested from more than one quarter that the only way of ensuring concentricity of the hob is to provide it with its own arbor, on which it would be finish-ground and from which it would never be removed. The Author takes this opportunity of expressing his strongest disapproval of this proposal. Hobs so mounted would be cumbersome and liable to damage, and with constant fitting of the tapered union the truth of the spindle, speaking in terms of fractions of 0.001 inch, would not be maintained. In any event, a profile cause of hobs running out of truth is eccentricity of the female taper and the provision of integral arbors would not overcome this defect. Other proposals have been put forward for driving the hobs with keys fitting into the end instead of the barrel of the hob, but all these suggested alternatives appear to avoid facing up to the principal causes of the present trouble, which can be rectified quite readily with due care and without essentially altering our standards. Apart from the cause already quoted, the principal source of trouble is slackness between the hob and its arbor and what can only be described as a miserable fit of the hob key, due essentially to the fact that despite all the care given to the manufacture and inspection of hobs, the keyways are machined before hardening and subsequent distortion is not corrected, as it should be, by grinding. Examination of hobs chosen at random will reveal keyways of many and almost grotesque shapes.

The first essential then is to make special provision for grinding these keyways to a fine tolerance. Hob shafts should be ground to about 0.0002 inch less in diameter than the hob bore, the spindle being cleaned with petrol and a drop of sperm oil applied before mounting the hob. The key and the keyway in the spindle should, of course, also be finished to a fine tolerance. The spindle should be checked for truth periodically and critically examined after any event such as the breakage of a roughing hob. The truth of the running of finishing hobs should be checked before and after each finishing cut.

Lack of Parallelism between Axis of Work and Travel of Hob Carriage. The necessity for achieving parallelism between the axis of the work and the travel of the hob carriage is very obvious and can only be obtained by the patient and unstinted application of skilled craftsmanship. Parallelism may be affected by warm or cold draughts playing on one side of the machine, and sudden changes of temperature may otherwise affect the accuracy of cutting; machines subject to such conditions should be carefully screened.

Improvement of Tooth Surfaces by Post-Hobbing Processes.

The various errors that in some degree are inseparable from gear hobbing introduce surface errors on the gear teeth that ultimately can only be eliminated by some other process. Current practice in such post-hobbing processes may be broadly classified as skilled handwork on the teeth, lapping with abrasive, and shaving. The effectiveness of each of these systems, so far as the various hobbing errors are concerned, is shown in analytical form in Table III.

The process which has frequently been adopted in marine practice in this country as necessity, or the desirability, has arisen has been handwork on the teeth—usually the pinion teeth—with possible subsequent light lapping, more appropriately termed "polishing," by means of a narrow lapping wheel running in mesh with the pinion against the action of a brake, and being slowly traversed across the face accompanied by the application of lapping compound. Handwork can obviously be effective in correcting for a divergence in helical angle of the meshing units, and

TABLE III. EFFECTIVENESS OF POST-HOBBING PROCESSES

Nature of Error	Finishing Process		
	Skilled Handwork	Shaving	Lapping with Suitable Technique
Defective helical angle	Effective	Not effective	Effective
Short-wave undulations	Impracticable	Effective	Effective
Long-wave undulations	Impracticable except for longest waves	Not effective	Effective
Pitch inaccuracy	Impracticable	Not effective	Not effective
Defective tooth profile	Can effect improvement	Effective	Can be effective, but if technique is not skilled the result can be the reverse
For highest quality gears the prime necessities of hobbing accuracy for the appropriate finishing process to be fully effective are:	No undulations No pitch errors Minimum profile error (Applied successively) No long undulations No pitch errors	No error in helical angle No long undulations No pitch errors	No pitch errors
Micro surface finish as produced by each process	Axial scoring eliminated by subsequent running	Polished	Matt surface consisting of radial scoring which does not appreciably reduce in service. Some abrasive embedded in surface. Pitch line polished
Comments	Pitch errors as hobbed must be an acceptable minimum under all circumstances		
	In the absence of long-wave hobbing undulations this combination process gives a seemingly ideal finish		If long-wave undulations exist they can be corrected by this process only and the lapped surface finish must be accepted

with skilled work under the guidance of marking, the tooth contour can also be improved to increase the radial depth of the contact band. Handwork, however, is not effective in the elimination of undulations on the teeth, particularly those of short wave, by reason of the impracticability of attempting to correct so large a proportion of the total bearing surface of the teeth.

The practice in the United States has differed from that in this country primarily as the skilled labor required for delicate handwork was not readily available, and latterly because the requirements for surface finish have become more stringent—necessitated possibly by the characteristic tendency of solid table cut gears to shriek. The practice which has been generally adopted there is that of lapping with full-width cast iron wheels. The usual procedure is for the wheel and pinion units to be lapped separately for a period of several hours, the lapping wheels either running against a brake load or being mounted on overhanging arms so that their own weight provides the necessary resistance; whichever method is used, it is customary for the lapping wheel to be in deep mesh so as to avoid the formation of a ridge on the teeth. The process is continued until contact over the full width of the helices is obtained, by which time the tooth thickness may have been reduced by several thousandths of an inch. Special technique is necessary in order to ensure that the region of the pitch line is not left proud, due to the absence of sliding contact at that line. Provision is made for this by adjusting the addendum distribution of the lapping wheel or by using a different flank angle on the lapping wheel from that of the finished teeth. The adoption of proper lapping speeds, tooth pressures and quality and consistency of lapping compound is essential to ensure that a minimum of abrasive action occurs in the regions where it is not required, and that it is the actual high spots that are progressively treated. By such means the lapping is suitably controlled to ensure that the tooth profile is not only maintained but perfected. Upon the completion of this process mating gears are mounted on suitable meshing frames, and are run in deep mesh, with the further application of lapping compound that may consist of medium fine carborundum or powdered glass. When satisfactory meshing is achieved the units are then run for a further period, and lubricated with clean oil to remove as much of the abrasive as possible.

Although not usual in marine work, similar practice has been employed in this country. It is effective in correcting for helical angle, both long- and short-wave undulations on the teeth, and also for tooth contour, but the latter only if the lapping technique is good and the amount of lapping not excessive. The surface of a lapped tooth does not present a pleasing appearance, being a matt finish with a semi-polished zone at the pitch line, and the fact that the minute score marks are in a radial direction mitigates against the possibility of their being minimized by the action of the teeth in service. Further, the introduction of abrasive in the manufacturing process is undesirable and calls for the greatest precautions in cleaning the units before final assembly.

The alternative process of shaving is now achieving considerable success in the United States, and under a similar operational technique will shortly be employed in this country. The gear to be shaved is mounted horizontally in bearings and is coupled direct to a motor drive. The shaving cutter, which is a special helical toothed wheel about 12 inches in diameter and 1 inch wide, is mounted on an idle spindle set in a carriage running on guides truly parallel to the axis of the wheel to be shaved. The teeth of the cutter present a series of axial cutting edges down each flank. The shaving cutter is run in deep mesh, with the work gear at a surface speed of about 400 feet per minute, and is slowly traversed across the gear face. It is essential that the cutter should bear on both sides of

the tooth gash simultaneously. The axis of the cutter is not parallel with the axis of the wheel, from which the process obtains its full name, "Crossed Axes Shaving," the object being that by such an arrangement the area of the contact zone is reduced and a high bearing pressure on the cutter teeth is thereby introduced without reducing the actual length of the zone; that must be kept as great as practicable. During the process the work is fed with a liberal supply of oil having a sulphur base. The procedure is essentially one of evening-out existing errors over a narrow zone, and is therefore virtually ineffective in the correction of long-wave undulations or of errors in helical angle, but it is ideal for the elimination of short-wave undulations and the correction of tooth contour, in which connection it seems to hold attractive possibilities for the perfection of tooth profile after any handwork that may have been necessitated by the correction of errors in helical angle. The surface finish that is given to the teeth is one of high polish, and is to all appearances very satisfactory.

None of the post-hobbing processes is effective in the correction of pitch errors in the teeth, and in this respect especially there is no substitute for a first-class hobbing machine. Shaving can only be fully effective if helical angle errors and long-wave undulations are held within a permissible fine limit, and although the former could, if necessary, be corrected by handwork before shaving, the latter in particular calls for a hobbing machine of the highest quality. In the absence of such perfection a highly finished tooth can only be achieved by lapping, with a consequent sacrifice in the microfinish of the tooth surface. A lapped gear cannot be shaved on account of the damage that would be caused to the cutters by residual abrasive. It is unquestionable that a higher standard of surface finish will permit of greater tooth loading, but to some extent the distinction in quality diminishes after a period of service and if the ultimate durability of the gears is not to suffer, the designer must take conservative advantage of the possibilities presented to him by improved production technique.

Gear Measurement.

Development in accuracy is to some extent governed by development in gear measuring apparatus and no paper devoted to gearing production would be complete without some reference to such equipment.

Comments on the principal features of measurement can best be made with reference to the errors listed in Table II.

Helical Angle. Measurement can now be derived by calculation from a direct measurement of axial pitch, employing a gauge developed at the N.P.L. by the late Dr. G. A. Tomlinson. In the application of this particular gauge two ball toes are set by length rod to the designed axial pitch of any desired number of teeth of the element to be measured, one of the toes being capable of axial adjustment by a micrometer head. The gauge is provided with a sensitive spirit level which is set centrally before commencing the measurement. The wheel to be measured is set with its spindle approximately horizontal and the gauge is applied into the bosom of the teeth at one side of the wheel, the micrometer head being adjusted until level is restored. The measurement is then repeated on the other side of the wheel and the micrometer values are averaged, the error in axial pitch being thereby defined. The measurement is obtained on the assumption that the helical angle is constant round the whole wheel, and this is normally justified within reasonable limits.

This overall measurement of helical angle is useful, but it is frequently of value to regard three individual components of the error and to measure these separately, namely, hob carriage movement, parallelism of hob slide with axis of work, and incidental errors of cutting. Fairly

definite measurements of the first two can be made, and these remain constant for any one machine over a considerable period. The measurement of incidental errors is made using the hobbing machine on which the gear was cut. A worm is mounted on the hob spindle in place of the hob and is held in contact with the ahead or astern tooth flanks by a spring, a dial gauge being mounted so as to bear on one abutment face of the worm. The hobbing machine is then set in motion so that only the spiralling gears are in action, the hob carriage moving and the hob rotating once for each axial pitch of the gear but the table remaining stationary. The effective difference between the known errors of the machine and the helical angle as cut is then indicated by the dial gauge readings, and any irregularity of cutting is thereby picked up immediately. This includes such errors as may have been caused by temperature effects or a temporary defect in the lubrication of the machine.

Reverting to the method of measuring hob carriage movement, it is essential that the measurements should be taken in way of the center line of the hob, or alternatively that simultaneous recordings be made at each side of the hob carriage and averaged, or as a further alternative, that single measurements taken off-center should be corrected by clinometer readings denoting the "rotation" of the carriage. Whether cumulative measurements recorded by the use of a short standard length rod and dial gauge are reliable, or whether the movement requires to be recorded on long smoke glass plates with the hob carriage in motion is a matter on which there is a difference of opinion.

Undulations. Measurements have depended entirely upon a recorder, designed by Dr. Tomlinson, consisting essentially of two ball toes set at a variable pitch and interspaced by a pivotted ball toe, that operates a recording arm scribing on a smoked glass disc which is rotated by the movement of the gauge along the tooth being measured; the fixed toes bear into the bosom of the tooth gash, thus registering the gauge, while the pivotted toe may be caused to bear on either flank as desired. Undulations are recorded with a magnification varying between (20×2) and (20×0) or 40 and 0, depending upon the relationship between the overall pitch p of the toes and the wavelength w of the undulation, the maximum corresponding with values of p equal to odd integer multiples of w and zero magnification with the even integer multiples. A certain amount of trial and error is involved in picking up the major undulations and considerable calculation may be necessary to analyze their fundamental cause, but the method is of extreme value in leading to the rectification of faults in the hobbing machine.

Circumferential Pitch. To obtain reliable cumulative measurements several requirements demand the utmost attention. In the case of a wheel, the periphery on which the gauge registers must be truly circular and concentric with the journals, a condition that can only be obtained with precision by grinding; also the radial face registering the gauge must be absolutely true, and this calls for the same procedure. In all cases the two toes of the gauge must bear on any particular tooth flank with identically coincident lines of contact, otherwise one toe will periodically ride on a microscopic high spot which the other misses, thereby falsifying the readings.

Tooth Thickness and Depth. Measurement either by Vernier gauge or comparator is susceptible to error due to lack of concentricity or circularity of the periphery of the gear, and the same precautions are called for as in the case of pitch measurements. Thickness should always be measured at or near the pitch line and in the case of all-addendum gears, where this is impossible, the readings are of little value unless corrected for the involuting of the tooth flank.

Tooth Profile. It is understood that various attempts have been made to devise a suitable gauge for this purpose, but a difficulty exists in that the measurements of error are of exceptionally fine degree and results up to the present have not justified the practical application of such a gauge.

LIGHT ALLOYS FOR MARINE ENGINES.

This paper by A. J. Murphy is reprinted from the March 1945 issue of the Transactions of the Marine Engineers.

Synopsis.

Since resistance to corrosion is of minor importance for materials to be used in the ship's engine room a wider range of light alloys is available than when external fittings are under consideration. For pistons the choice of Y-alloy is confirmed by recent data on creep strength and on performance under combinations of steady and alternating stresses at temperatures up to 300 degree C. When alloys of low coefficient of thermal expansion are being selected for pistons their thermal conductivity must be taken into account as a factor influencing the actual diameter attained under running conditions. There is a prospect that a cylinder head made of a corrosion-resistant aluminium-magnesium alloy might permit direct cooling by sea water.

Wrought alloys of aluminium find application in marine engines for connecting rods, rockers, operating rods, fan blades and covers. Modern means of controlling foundry technique and inspection have made it possible to use light alloy castings for stressed components in aeronautical engineering and it is reasonable to apply this experience in marine engine design. Evidence is given regarding the tensile and fatigue strength of specimens taken from aluminium- and magnesium-alloy castings. Light alloys generally are not more notch-sensitive than steels in dynamic loading. Internal stresses in light alloy castings can be largely relieved by heating at about 300 degree C., but when strong heat treated alloys are involved the effect on the mechanical properties has to be studied. In Y-alloy, quenching in boiling water introduces lower internal stresses than quenching in oil.

The great increase in quantities of light alloys made and used during the war years has provoked much speculation on the extent to which these materials will be called for when the abnormal consumption in armaments has ceased. From the beginning of hostilities, and especially since 1940 until the present time, immediate war needs have restricted the applications of light alloys practically to those fields where no alternative was possible. Aircraft and incendiary bombs have therefore been the destination of most of the aluminium and magnesium produced during the last five or six years. During this prolonged phase of restriction, knowledge of the properties of light alloys and of the methods whereby they may be manipulated has advanced at a steady, if not startling, rate. Those who come to put these materials to new uses when difficulties of supply have disappeared will therefore have the benefit of an accumulated stock of data which was not available in 1939.

Ships have not been overlooked by those studying post-war applications of light alloys. The case for aluminium alloys in the structure of ships has been ably presented recently by a number of writers who have been sufficiently bold to invite comparisons with steel, which has had the field to itself for so long. It is not the author's intention to deal with this aspect, but to confine his remarks to the part which light alloys may play in the marine engine and its ancillaries.

It is advisable at this stage to explain the angle from which the subject will be approached. The author is not an engineer and is certainly not competent to present to this Institute a discourse on the full consequences as regards mechanical performance which would attend the use of light alloys for a particular component. He will therefore be content if he can discharge the less ambitious task of outlining some of the metallurgical considerations which appear to be relevant. The stream of facts and figures in publications on light alloys emanating from research institutions, private companies and trade associations is as accessible to the marine engineer as it is to the metallurgist, but the metallurgist may be of service in indicating how the various alloys fall into classes, what are the basic characteristics of these classes and what are the factors which should influence a selection from them.

In the category of light alloys it is desired to include both the aluminium-base and the magnesium-base series. It is perhaps a consequence of so much of the discussion on marine applications of light alloys being concerned with the hull and general structure, where resistance to corrosion is the primary consideration, that the service which magnesium-base alloys might render in the engine room appears to have been overlooked. Although the two classes share the feature of low density it is necessary to bear in mind that in other respects there are important differences.

What are the characteristics of light alloys which may make them of interest to the designer, constructor and user of marine engines, and what are their limitations which must also be taken into account? In engine work the lightness of the alloys of aluminium and magnesium is their most important quality. All the aluminium-base alloys which have mechanical properties of interest for use in engineering components have specific gravities lying between 2.6 and 3.0, equivalent to densities of .0937 to .108 pound per cubic inch, or 161.9 to 186.6 pound per cubic foot. The magnesium-base alloys fall within a narrower range of 1.77 to 1.82, i.e., .064 to .066 pound per cubic inch or 110.24 to 113.35 pound per cubic foot. A comparison of these with the corresponding values for mild steel, grey cast iron and gunmetal is as follows:—

TABLE I.

Material.	Specific Gravity.	Density lb./cu. in.	Density lb./cu. ft.
Magnesium-base alloys	1.8	.065	112.0
Aluminium-base alloys	2.8	.101	174.4
Cast iron	7.5	.270	466.6
Mild Steel	7.8	.281	485.6
Gunmetal	8.8	.317	547.8

The consequences which follow the reduction in weight of an engine component depend upon whether its service is static or dynamic. In the first category there are parts such as crankcases, bedplates, gearcase covers and inspection covers. A saving in the weight of these components, such as is achieved by changing from a ferrous material to light alloy, lightens the whole engine to an extent which may be significant or not according to the nature of the vessel Thus in a high speed motor boat a light weight engine gives greater scope to the designer in the location of the engine in the vessel and simplifies the problem of strengthening the hull and framing to support the engine. In the large ship it is not so obvious that these considerations may have similar force, although they come to light occasionally. An example was in one of the German pocket battleships, of Washington Treaty days, in which main engine bedplates

were reported to have been made of magnesium alloy as a contribution to the all round saving of weight.

Weight saving might be expected to be of direct appeal in the case of parts such as inspection doors and gear covers on large engines. In the restricted space of a ship's engine room it must be a great boon if a cover which has to be lifted rather frequently and which normally requires two men, or mechanical assistance, to move it can be so lightened in weight that lifting becomes a one man job.

Pistons.

While the foregoing points have special reference to marine engines, the influence of lightness in reciprocating parts is similar in all types of engine. In internal combustion engines the reciprocating component which has received most attention from this point of view is undoubtedly the piston. The author assumes that all interested in this subject will have read the paper by Mortimer and Paige on "Light Alloys in Heavy Oil Engines" presented to the Diesel Engine Users' Association in January, 1944, and he will not presume to go over familiar ground again. Mortimer and Paige considered light alloy pistons under the headings of alloy composition, design, clearances, heat treatment, and stress relief. Their specification for a cast light alloy piston can be summarized, with reference to these features, as: Y-alloy type or Lo-ex, simplest possible design with a minimum of ribs, clearances at the top edge of 0.008 inch per inch of diameter for Y-alloy and 0.006 inch for Lo-ex; heat-treatment calculated to give the greatest freedom from stress and the greatest stability at operating temperatures rather than the highest possible strength.

These authors were justifiably at some pains to warn engineers against selecting alloys for light alloy pistons on the basis of the tensile strength at ordinary temperatures or the strength recorded in short time tests at elevated temperatures. They did not, however, continue the discussion to the point of indicating the criteria by which a piston alloy would be selected if experience of its behavior in service were not available. It may be useful to consider this a little further.

As regards the first point—the strength at the ordinary temperature—the situation is fairly clear. It is the strength at the maximum temperature attained in operation which is significant, and without some knowledge of the rate at which the strength falls off with rising temperature, data relating to the mechanical properties at room temperature are irrelevant.

On the second point it is interesting historically that Y-alloy, which still holds its place in the front rank of piston materials, was selected for that purpose on the basis of short time tensile tests at elevated temperatures. In the original investigation at the National Physical Laboratory the test pieces were held at the temperature of test for only 30 minutes before loading to fracture. This was sufficient to bring out the superiority of Y-alloy, containing 4 per cent of copper, 2 per cent of nickel and 1½ per cent of magnesium over the Duralumin, nickel-free type of alloy and the straight aluminium-copper series such as L. 8 alloy. It is not the author's intention, however, to suggest that the strength in short time tensile tests ought now to be accepted as a reliable criterion for service at high temperature; at the best such tests can only be regarded as indicating whether a material merits further consideration for such an application.

When we enquire what type of test other than short time tests would bring out most effectively the characteristics needed for a component operating under reciprocating stresses at elevated temperatures, we think of the two ways in which failure could occur by a gradual process under those conditions, namely, through fatigue on account of alternating stresses or through creep under a steady stress at the temperature of operation. The author believes it is the general view, and one which seems natural to him, that to take into consideration only the creep resistance would be

TABLE II.

CREEP AND FATIGUE STRENGTH AT ELEVATED TEMPERATURES.

Material.	Treatment.	Mechanical Properties.					
		20° C.		200° C.†		300° C.‡	
		Ultimate tensile stress, kg./sq. mm.	Elong'n. per cent.	Limiting creep stress,* kg./sq. mm.	Fatigue stress, 10 million cycles, kg./sq. mm.	Limiting creep stress,* kg./sq. mm.	Fatigue stress, 10 million cycles, kg./sq. mm.
(A) Aluminium-copper-nickel (Y-alloy type)	Hot pressed heat-treated	34·5 (21·89)	17	14 (8·89)	±10 (6·35)	3 (1·90)	±4·6 (2·92)
(B) Aluminium-silicon-copper (Duralumin type)	Hot pressed, heat-treated	30·2 (19·15)	15	12 (7·62)	±6·2 (3·94)	<2 (1·27)	±4·0 (2·54)
(C) Aluminium-magnesium (Hydronalium type)	Hot pressed	26·2 (16·54)	20	<3 (1·90)	±5·8 (3·68)	<1 (0·63)	±3·8 (2·41)

Values in brackets represent conversion to tons per square inch.

† Materials A and B annealed 48 hours at 200° C. before test.
Material C " 1 hour " " " " "

‡ Materials A and B " 24 hours " 300° " " "
Material C " 1 hour " " " " "

* Stress producing a rate of creep of 10×10^{-4} per cent. per hour between the 25th and 35th hours of loading.

an inadequate basis for the selection of a material for pistons. The feature of alternating stress appears to be inseparable from the service of a piston, and the most important mechanical property in this application is therefore the endurance under fatigue at the highest temperature attained by stressed parts of the piston in operation. Nevertheless, if the mean stress of the fatigue cycle is of appreciable magnitude, and especially if it is a tensile stress, we have the condition of a steady creep-inducing load operating at the same time as the fatiguing forces.

Published data on the creep behavior and the fatigue properties of light alloys at elevated temperatures are still not numerous, and variations in the testing conditions, e.g., in the number of reversals of stress for which the endurance value is reported, make it difficult to compare the results of different investigators. From the information which is available, however, it is clear that among all light alloys in regular use those of the Y-alloy type are the best in respect of both fatigue strength at temperature around 300 degrees C. and also in creep resistance.

Two sets of fatigue endurance curves, giving a comparison between Y-alloy, an alloy of Duralumin type and Hydronalium, at temperatures of 200 degrees C. and 300 degrees C. are reproduced in Figure 1*. The observations were only continued to 10-million reversals of stress. In general it is desirable to base fatigue data on aluminium alloys on a large number of cycles of stress, even as high as 500 millions. In the tests reported here, however, the reference to a smaller number of reversals appears to be justified by the distinct flattening of the curves before 10 million cycles have been attained It is evident that Y-alloy has a greater resistance to alternating stress at 200 degrees C. and 300 degrees C. than the other two materials, while there is little difference between these two.

The published information on creep of aluminium alloys is more extensive. Typical data are shown in Figure 2. The principal feature again is the outstanding merit of Y-alloy. It is also to be noted that the cast form is distinctly more resistant to creep than the wrought at temperatures of 200 and 300 degrees C., both being initially heat treated to possess the highest strength at ordinary temperatures. The superiority of the cast form in creep resistance is fairly general.

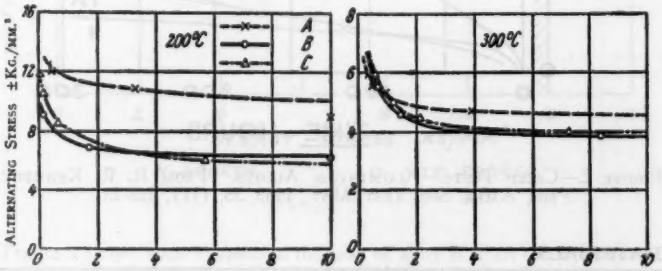


FIGURE 1.—ENDURANCE (MILLION CYCLES).

The account of German work published two years ago from which Figure 1 is taken is interesting for the results it gives relating to the effects of combined creep and fatigue at elevated temperatures. The investigators studied three aluminium alloys of the Y-alloy, Duralumin

* Siebel, Steurer and Stähli. *Z-fur Metallkunde*, July, 1942, *34*(7), pp. 145-150.

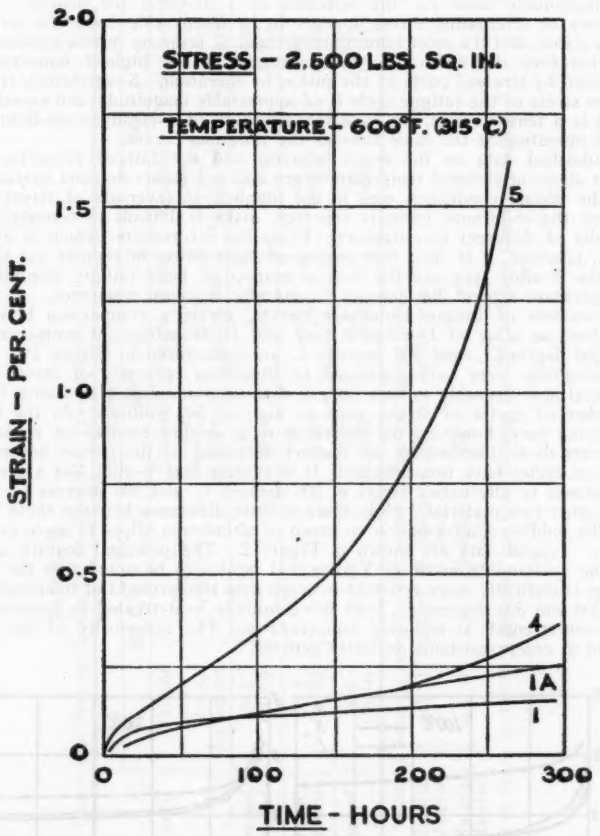


FIGURE 2.—CREEP TESTS.—ALUMINIUM ALLOYS. FROM R. R. KENNEDY, PROC. AMER. SOC. TEST. MAT., 1935, 35, (II), 218-231.

MATERIALS:

		Compositions per cent. Aluminium Balance. Copper.	Iron.	Silicon.	Magnesium.	Nickel.	Manganese.
1	Cast	3.90	0.38	0.21	1.60	2.22	
1A	Wrought	3.91	0.35	0.13	1.68	2.22	
4	Cast	4.29	0.50	0.66			
5	Wrought	0.90	0.78	12.64	0.96	0.91	0.02

All materials stabilized 20 hours at 315° C., after heat treatment.

and Hydronalium types. Their objective, in essence, was to determine the combinations of continuous and vibratory stresses which the materials could withstand without either fracturing or extending by more than a certain specified amount. The experiments were conducted at 200 degrees C. and 300 degrees C. in a machine which imposed alternating tension and compression on the specimen while it was stressed in tension to the desired degree. In other words, in the fatigue test the mean value of the range of stress could be varied from zero to a positive tensile value. The specimens were cut from press-forged pistons.

Table II gives the tensile strength and ductility of each material at the ordinary temperature, together with the limiting creep stress and the fatigue stress on a basis of 10-million cycles at 200 degrees C. and 300 degrees C.

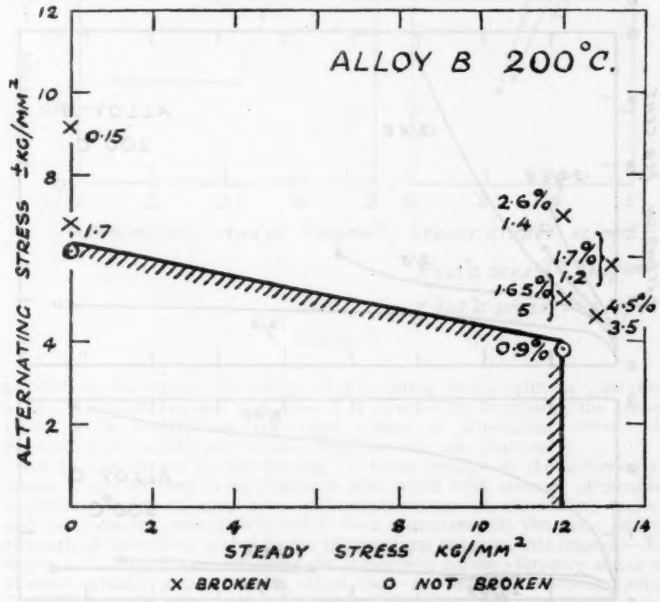


FIGURE 3.

Figure 3 shows what happens in the case of alloy B when the alternating stress is imposed upon a steady stress at 200 degrees C. With the range of alternating stress as ordinate and the value of steady stress as abscissa, the area below the cross hatched boundary defines the combinations of alternating and steady stress which can be imposed at 200 degrees C. or 300 degrees C. without causing fracture or extension beyond 1 per cent in 45 hours. The figures shown against the plotted points indicate the extension and the endurance in million cycles up to fracture. Figure 4 shows the progress of creep under the various combinations of stress; the values of the steady stress and of the semi-range of alternating stress are denoted against each curve. Figure 5 illustrates how the extension is

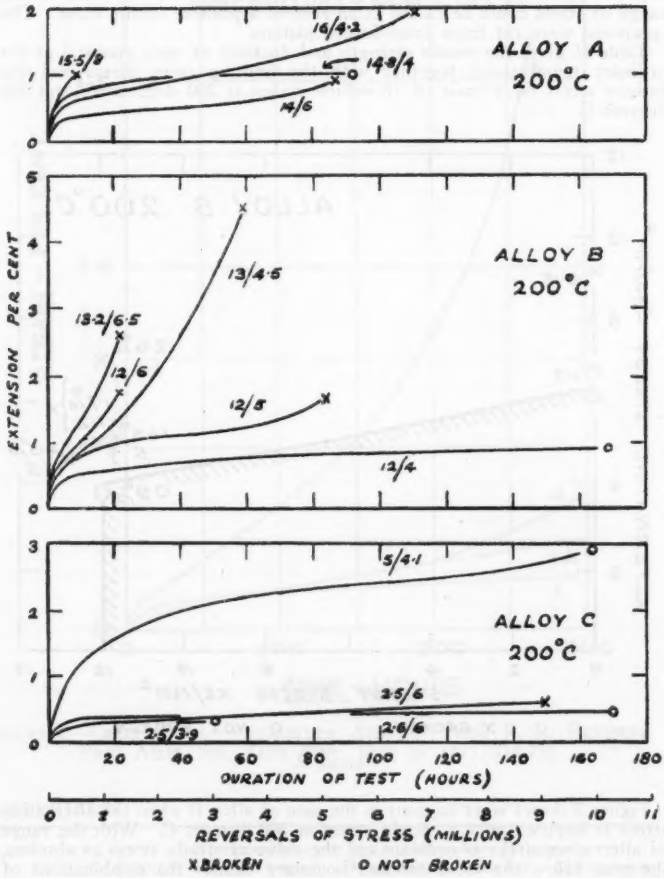

ALLOY A
200°C
16/4·2
15·5/8
14·8/4
14/6
ALLOY B
200°C
13/4·6
13·2/6·5
12/6
12/5
12/4
ALLOY C
200°C
5/4·1
2·5/6
2·5/3·9
2·6/6
EXTENSION PER CENT
DURATION OF TEST (HOURS)
REVERSALS OF STRESS (MILLIONS)
X BROKEN
O NOT BROKEN

FIGURE 4.

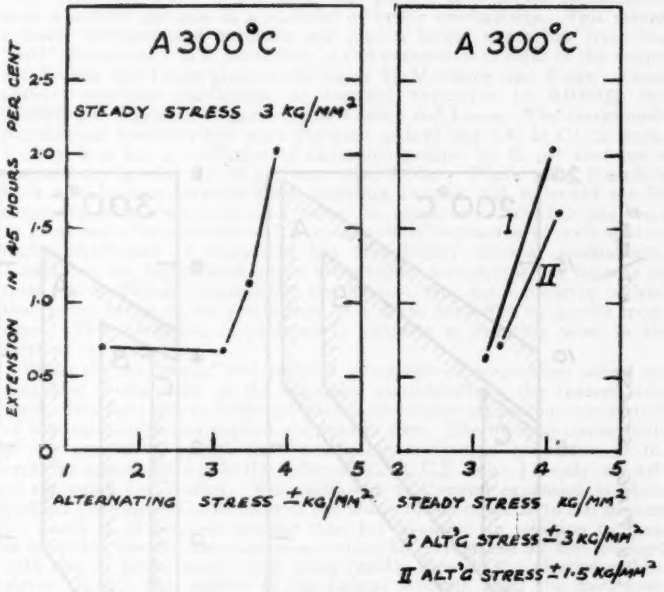


FIGURE 5.

affected by increasing the range of alternating stress without alteration of the steady stress, and also how it is affected by increasing the steady stress while maintaining the same range of alternating stress. In Figure 6 these results are brought together into one diagram.

As an illustration of the bearing of these results on the selection of alloys, it can be seen from Figure 6 that while with stresses alternating symmetrically about zero (i.e. with equal values of alternating tension and compression) materials B and C have approximately the same fatigue strength at 10-million cycles in the temperature range of 200 degrees—300 degrees C., alloy B can withstand the imposition on the vibratory stress of a much greater steady tensile stress than can alloy C without failing through fracture or excessive extension. Another interesting observation is that in all three alloys, with the steady stress at the limiting creep value, an additional vibratory stress can be imposed without causing excessive deformation.

The collective information supplied by these results confirms in a striking manner the all-round superiority of the Y-alloy type of aluminium alloy for service in pistons, cylinders and cylinder heads. It is indeed remarkable testimony to the accurate judgment of Walter Rosenhain that the alloy which he first proposed 27 years ago should have performed so brilliantly under the more exacting conditions of testing which have since been developed.

Thermal conductivity and thermal expansion are physical properties of first rate importance in piston materials. A relatively low conductivity is a disadvantage in more than one respect. By retarding the flow of heat

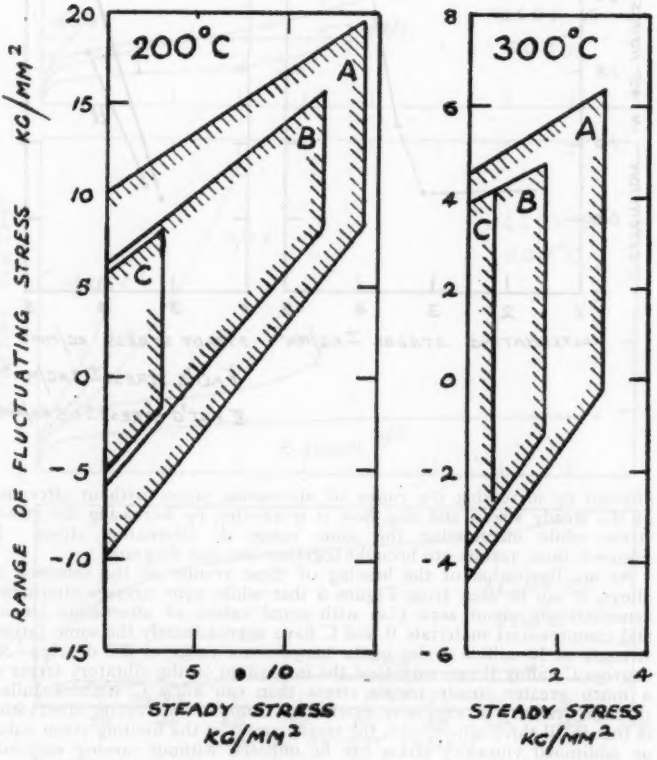

200°C
300°C
A
B
C
RANGE OF FLUCTUATING STRESS KG/MM²
STEADY STRESS KG/MM²
STEADY STRESS KG/MM²
20
15
10
5
0
-5
-10
-15
8
6
4
2
0
-2
-4
-6
5
10
2
4

FIGURE 6.

away from the crown it maintains that portion at a higher temperature than would be the case in a material of better conductivity. This means a lower strength and hardness and also a larger expansion from the "cold" dimensions. It is interesting in this connection to refer to the values for Y-alloy and Lo-ex given in the paper by Mortimer and Paige. These authors attribute coefficients of thermal expansion of 0.000022 and 0.000019 per degree C. respectively to Y-alloy and Lo-ex. The corresponding thermal conductivities were reported as 0.40 and 0.32 in C.G.S. units. Y-alloy thus has a coefficient of expansion greater by 16 per cent but a conductivity greater by 25 per cent than Lo-ex. There must therefore be a nice balance between these opposing factors, and it would not be surprising if in some designs of piston the expansion at the top ring were as great with the material of low coefficient of expansion as with that of higher coefficient of expansion but also higher thermal conductivity. Sometimes the high-silicon alloys are adopted for pistons not because of their low coefficient of expansion but because they are considered to lend themselves better to the production of a thick, hard film by anodic treatment. This hard film is particularly valuable in reducing wear in the skirt of the piston.

When the mechanical and physical properties of magnesium alloys are examined in the light of the foregoing considerations, the reasons why these ultra-light alloys have not found successful application for pistons of internal combustion engines are readily seen. The thermal conductivity of the cast alloys which have reasonable mechanical properties at the ordinary temperature is in the order of 0.20 C.G.S. units, i.e. only one-half of the value for Y-alloy. The coefficient of thermal expansion is about 0.000027 per degree C. for most of the alloys in the range 20 to 200 degrees C., which is 25 per cent greater than for Y-alloy. In addition to these unfavorable factors the magnesium alloys in current use all lose strength with rise of temperature much more rapidly than do the aluminium-base piston alloys. This applies to the fatigue strength also; the most commonly used casting alloys of magnesium have fatigue strength between ± 5 and ± 7 tons per square inch at 20 degrees C., falling to ± 2.5 to ± 3.5 tons per square inch at 150 degrees C., a decrease of 50 per cent. In Y-alloy the corresponding fall is nearer to 20 per cent.

Cylinder Heads.

The selection of a light alloy for a cylinder head of an internal combustion marine engine must be guided by the same main principles as those which apply when the service is on land or in the air. Good thermal conductivity and adequate strength at the operating temperature are the main requirements. In liquid-cooled cast heads there is the complication in marine engines that the most readily available cooling medium, sea water, is corrosive towards the aluminium alloys containing copper or nickel or both, most commonly used where cooling is by fresh water. The difficulty is overcome by using a closed circuit of fresh water circulating through the cylinder head, with a heat exchanger made of non-corrodible metal cooling the fresh water through the agency of sea-water. This would appear to be rather a troublesome device and the author has wondered whether, if it has not already been done, it would not be worth while to make a trial with a head cast in a light alloy offering at least a prospect of being able to withstand the corrosive action of hot seawater —for instance one of the aluminium-magnesium alloys, such as Birmabright, containing about 5 per cent magnesium. These have been used successfully for sea-water cooled exhaust manifolds, which present conditions tending to promote corrosion similar to those prevailing in a cylinder head, although in the latter the temperature is higher and the stresses are greater. An observation relating to a German air-cooled aero

engine cylinder head which fell into our hands could be quoted to encourage the suggested experiment. This cylinder head, in a B.M.W. (801) engine from a Dornier 217 aeroplane, was a casting having the composition (in round figures) :—

Magnesium	..	5 per cent
Silicon	.. ..	1 per cent
Titanium	.. ..	0.1 per cent
Aluminium and impurities	..	remainder.

Examination of the microstructure indicated that the casting had been heat-treated, which is known to improve still further the corrosion resistance of this alloy. The mechanical properties on a sand cast test-bar, viz :—

Proof stress (0.2 per cent)	6 tons per square inch.
Ultimate tensile stress ..	11-12 tons per square inch.
Elongation	5-2 per cent
Brinell hardness (5/250/30)	60-70.
Fatigue strength	±3.8 tons per square inch.

are very similar to those of the alloys named by Mortimer and Paige as suitable for cylinder heads (Birmasil Special, Alpax Beta, R.R. 50 and Ceralumin B).

It is not clear why the Germans should have used this alloy for an air-cooled cylinder head—unless it was that its foundry properties made it specially suitable for producing the thin fins—but it can be assumed that it had probably proved to be generally satisfactory at least as regards mechanical strength.

If the proposed trial were made it would be essential to avoid contact between copper alloys and the light alloy in the cooling sea-water, as this would cause violent galvanic corrosion of the aluminium alloy. There would not appear to be any great difficulty in arranging this, but it might be useful to mention that turnings or chips of brass or bronze accidentally finding their way into the water passages of the castings would have very harmful corrosive effects.

Corrosion.

Having been brought to the subject of corrosion through the discussion of water cooled, light alloy, cylinder heads, we may consider this an opportune point at which to refer to the question in broader terms.

It is the author's impression that marine engineers have been diverted from a full appreciation of the service which they might gain from light alloys in the engine room, where the prevailing conditions do not cause severe corrosion, by discussions and experiences of the use of light alloys for ships' hulls, superstructures, and deck fittings which are exposed to seawater and seawater spray and which demand as a first requirement high resistance to corrosion. The author is aware that aluminium alloys are already used for such engine room components as crankcase inspection doors, gear casings and covers, which are exposed to the atmosphere of the engine room. The alloy most commonly used for these parts, however, is aluminium-silicon, which is notable for its resistance to corrosion; the success of this application therefore could not be held in itself to demonstrate that corrosive influences are absent from a ship's engine room and could not be used as an argument for permitting the use in and around marine engines of light alloys less resistant to corrosion. The required evidence is provided by steel. Hand rails, floor plates, brackets and levers of bare steel in the engine room are not found to suffer rapid deterioration through corrosion, and on those parts which are wiped

regularly with an oily rag a bright surface is maintained without much difficulty. These observations lead to the conclusion that, except for parts in contact with seawater as a cooling medium, the selection of light alloys for marine engine components need not be influenced by considerations of resistance to corrosion. If this view is accepted, a wide range of alloys, magnesium-base as well as aluminium-base, cast and wrought, is available to the designer and builder of marine engines.

Mechanical Properties.

What are the applications for which their mechanical characteristics make light alloys suitable?

To answer this question adequately would be a large undertaking since it would be necessary to deal with a score or more of alloys, in the forms of castings, forgings, extrusions and rolled products and with the effects of heat-treatment, hot working and cold working. To avoid merely producing a catalogue of mechanical properties it is necessary to make a selection from these numerous divisions of the subject. It is proposed therefore to touch on wrought light alloys only briefly by referring to instances where they have already found application in the marine engine.

Wrought Light Alloys.

The alloys of aluminium which are used as forgings are generally of the Y-alloy type, which includes the R.R. 56 and R.R. 59 alloys, Ceralumin F and Ceralumin 22. These have tensile strengths of 27 to 30 tons per square inch at the ordinary temperature and retain their mechanical properties rather well at elevated temperatures. The alloys of the Duralumin group have similar, or sometimes superior, strength at the ordinary temperature, but they lose strength more rapidly with rise of temperature and are therefore not suitable for pistons and cylinder heads.

The principal applications of aluminium alloy forgings in the field in which we are interested are for pistons, connecting rods, rockers and rocker-operating rods, fan blades, small caps and covers. Forged pistons are confined to the smaller engines, especially those of aeronautical type. Light alloy connecting rods have proved valuable in reducing inertia. The low modulus of elasticity of aluminium alloys, 10 million pounds per square inch, as compared with 30 million pounds for steel, necessitates increasing the depth of section to increase the moment of inertia, if equivalent stiffness to that of a steel rod is to be attained, but in spite of this a saving in weight of about 50 per cent can be effected. The higher coefficient of thermal expansion of the aluminium alloys is an unfavorable factor, where a reduction in the volume of the combustion space as the connecting rod warms up cannot be tolerated. On the other hand, the more massive section of the light alloy rod may prevent any objectionable expansion by diminishing the rise of temperature. It is possible to run hardened gudgeon pins and even crankpins directly in forged aluminum alloy connecting rods, but separate inserted bearings are preferable, if only for the sake of confining the damage to an easily replaceable component in the event of a seizure through some cause such as a temporary failure of lubrication.

Rolled and extruded aluminium alloys have been used for chequer plates, handrails, fuel pipes and fuel tanks. The alloys containing 3 to 5 per cent magnesium have been found suitable for these applications and have the advantage over stronger alloy of being easily welded.

Cast Light Alloys.

Since the idea of using light alloy castings for stressed components may not yet have gained acceptance, or even consideration, by some marine engineers it might serve a useful purpose to deal at somewhat greater length with this aspect of the subject.

If we were thinking only of casings and covers which have to carry practically no load other than their own weight, little interest would be attached to the strength of the material, except perhaps indirectly in instances where for example the effects of imposed vibration and shocks may have to be taken into account. The author wishes to carry the examination of light alloy castings further than this, however, and to discuss their suitability for stressed components. Although his case may be illustrated by examples of engine parts made as light alloy castings, it is not proposed to attempt to make a list of such items for which light alloys might present special advantages. Marine engineers are much more competent than is the author to do this, when once they are convinced that they have the data to define the essential characteristics of these materials.

The first broad indication of the mechanical properties of these, as of other cast alloys, is to be gained from a sample cast in a standard form as provided for in one or other of the specifications, e.g., of the British Standards Institution or Ministry of Aircraft Production (Directorate of Technical Development). It is not considered necessary at this stage to take up much time in discussing the limitations of these test bars. Their purpose is to provide evidence as to the quality of the metal which the foundryman has used, but as they are not subjected to the same conditions of running and feeding, nor to the same effects of mass and section, it is clear that there is no justification for assuming that the properties found in various parts of the casting will be the same as those recorded in the test bar; they may be greater or smaller according to the circumstances of the solidification of the metal in the mould. Test samples cast integrally with the casting give no better guide, since the conditions of solidification in the attached test bar cannot be representative of the conditions in other parts of the casting. Fortunately, experience gained during recent years in the application of light alloy castings for stressed parts, especially in aircraft, has shown how a basis for design of greater value than any standard test bar data can be established.

Although it was in the cast form that non-ferrous metals found their earliest application in engineering and military arts, as soon as the equipment necessary to produce forgings, and then to roll and extrude, became available the wrought material was preferred for parts which were of special importance. Forgings gained their place because they were more reliable. Castings were less consistent in quality and there were no means of determining with certainty whether some internal defect was present in the particular casting which it was desired to use. On the other hand it was always true that castings possessed advantages over forgings in some respects. They could be produced with simpler equipment, patterns for castings could be prepared at a fraction of the cost and in much less time than dies for forgings. If the comparison were to be made with forgings produced without dies then the casting would leave less machining to be done in shaping the finished article. A modification found necessary after series production had begun could be introduced with less delay and dislocation in the case of castings. An advantage of castings more fully appreciated in recent years has been the comparative freedom which they give the designer from the limitations of shape and form imposed by other methods of manufacture.

The renascence, for it is little less, of castings for critical engineering service has been made possible by new conceptions of technical control in the foundry and by improved means of inspection, especially by the use of X-rays.

The designer wishes to know the properties of his materials in the forms in which he proposes to use them; variation in the mechanical properties throughout a casting will be greater or smaller as the complexity of form and the variations in section are greater or smaller.

FIGURE 7.

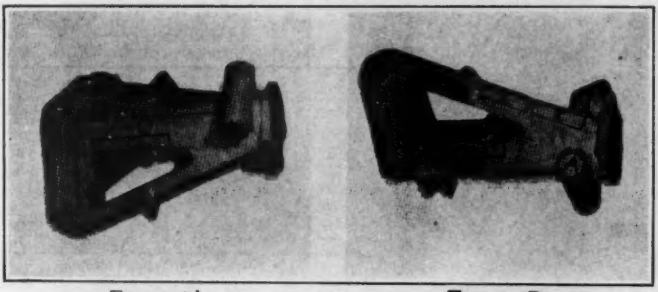


FIGURE 8A. FIGURE 8B.

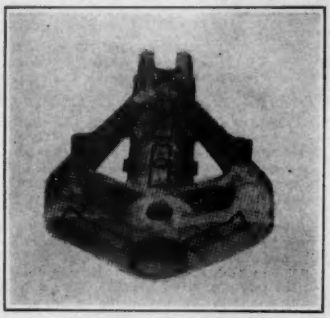


FIGURE 8C.

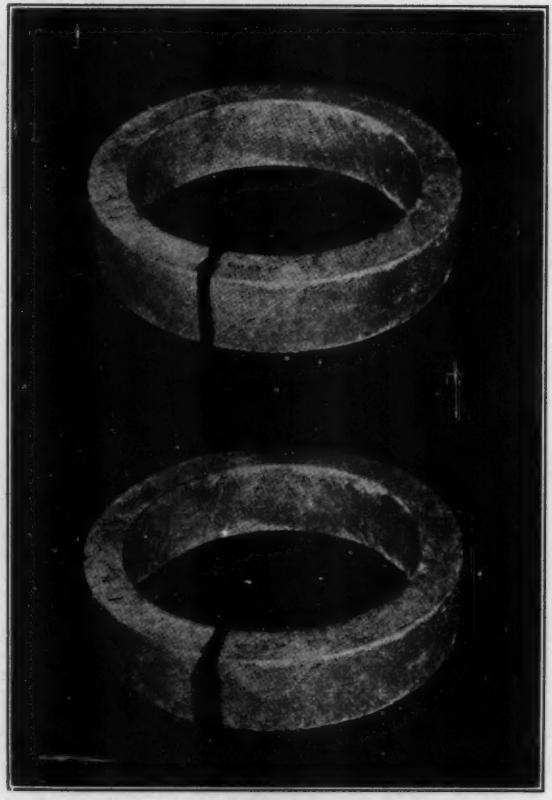


FIGURE 12.

TABLE III.

TAIL WHEEL FORK P. 313. MATERIAL: ALUMINIUM ALLOY ASM.

Test piece mark.	Diameter of test piece.	0.1% Proof stress tons per sq. in.	Ultimate stress, tons per sq. in.	Elongation % on 1.5 in.
F A	.564″	11.00	16.60	7.5
F B	.564″	9.60	15.76	7.5
F C	.564″	9.36	15.20	7.5
F D	.564″	9.84	13.20	4.0
F F	.564″	9.60	14.88	6.0
F H	.564″	9.32	13.52	4.0
F K	.564″	9.40	13.88	4.0
	Minimum	9.32	13.20	4.0
	Maximum	11.00	16.60	7.5
	Average	9.73	14.72	5.8

TABLE IV.

TAIL WHEEL FORK P. 574. MATERIAL: ALUMINIUM ALLOY ASM.

Test piece mark.	Diameter of test piece.	0.1% Proof stress tons per sq. in.	Ultimate stress, tons per sq. in.	Elongation, per cent.
F A	.799″	12.40	16.84	4.0*
F B	.565″	11.20	16.00	4.0†
F C	.800″	10.98	15.55	6.0†
F D	.564″	11.40	13.64	2.0*
F E	.799″	11.90	15.12	3.0*
F F	.564″	11.76	17.60	6.0*
F G	.565″	11.92	17.68	5.0*
F H	.701″	—	12.85	2.5*
F J	.798″	11.58	14.40	3.5*
F K	.564″	11.72	17.00	5.0*
F L	.565″	11.64	15.64	3.5*
F M	.565″	—	19.00	8.5‡
F N	.564″	—	16.84	7.0‡
	Minimum	10.98	12.85	2.0
	Maximum	12.40	19.00	8.5
	Average	11.65	15.78	4.6

* Gauge length 1.5″ † Gauge length 1.0″ ‡ Gauge length 0.5″

In light alloys the most important factors affecting the distribution of properties are the degree of local unsoundness and the thickness of sections. By co-operation between designer and founder the influence of these factors can be studied exhaustively in one pattern of casting typical of a class. Once the foundry technique has become established, the corresponding mechanical properties in the important parts of the casting are known, and by the application of radiological examination, coupled if necessary with occasional destructive tests, the consistency of subsequent production can be assured. Not the least useful contribution which the founder brings to the consultation is his knowledge of the casting characteristics, or "foundry qualities" of the different alloys available, since one type of alloy by its constitution will be more seriously affected than another by unfavorable circumstances such as thinness of section or restricted contraction. Authoritative information on the subject has recently become available as the result of a co-operative investigation by the Light Metal Founders' Association[1].

Because aeronautical engineering has always been faced with the inescapable necessity for saving weight, it is in the production of aero engines and aircraft that the application of light alloy castings for stressed components has made the greatest progress. For this reason the author turns to aircraft castings in order to illustrate the standard of quality attainable by the procedure outlined above. Although the examples quoted are drawn from a different branch of engineering, it will not be difficult to recognize the analogies existing between these and certain components in marine engineering construction.

Tensile Properties of Light Alloy Castings.

The first evidence which the author wishes to cite is a report of strength investigations carried out on a tail-wheel landing fork, having an overall length of 13½ inches and weighing in aluminum alloy 11¼ pounds. The form of this casting is seen in Figure 7.

The alloy to which the first two sets of values apply was one designed to provide a combination of good casting quality with adequate mechanical strength. Other alloys could have been chosen to give higher strength and ductility in a separately cast standard test bar, but because of their inferior casting qualities the risk of harmful defects in the casting would have been greater. The specification properties of the alloy used were

0.1 per cent proof stress	..	Not less than 10 tons per square inch.
Ultimate tensile stress	..	Not less than 15 tons per square inch.
Elongation	..	Not less than 4 per cent.

Table III shows the results of mechanical tests on specimens cut from the positions indicated at one stage in the development of the foundry technique. Table IV shows a later stage.

An indication of the range of strength values obtained from a much harder, less ductile, alloy is given by the results in Table V.

In this case the specification values were:

0.1 per cent proof stress	Not less than 17.5 tons per square inch.
Ultimate tensile stress	Not less than 18 tons per square inch.
Elongation	Not specified.

It happens that the same part is made also in magnesium alloy Elektron A 8 to specification D.T.D. 289, which in a standard test bar calls for an ultimate tensile stress not less than 13 tons per square inch and an elongation not less than 6 per cent. The proof stress expected in this bar

[1] Light Alloy Castings; Some Notes on the Relationship of the Material and Design. Light Metal Founders' Association, 25, Bennett's Hill, Birmingham, 2.

is 5 tons per square inch. Table VI gives the results of tensile tests on specimens taken from the same positions as were illustrated for the aluminum alloy castings. It will be seen that although the weight of the casting in Elektron is only 7½ pounds as compared with the 11¼ pounds for aluminum alloy the average strength in the lighter material is only slightly lower (15.32 tons per square inch) than that in A.S.M. aluminum

TABLE V.

TAIL WHEEL FORK. MATERIAL: ALUMINIUM ALLOY—D.T.D. 255.

Diameter.	Mark.	0.1% Proof stress, tons per sq. in.	Ultimate stress, tons per sq. in.	Elongation % on 1.5 in.*
.564″	A	19.6	21.9	0.8
.563″	B	19.6	20.4	0.8
.564″	C	—	17.8	0.5
.798″	D	—	23.6	1.3
.564″	F	—	24.8	1.07
.749″	H	—	21.6	Nil†
.864″	K	20.4	24.4	1.3
	Minimum	19.6	17.8	0.5
	Maximum	20.4	24.8	1.3
	Average	19.9	22.1	1.0

* Non-standard gauge length. † Broke outside gauge marks.

TABLE VI.

TAIL WHEEL FORK. MATERIAL: ELEKTRON A. 8—D.T.D. 289

Bar.	Width.	Thick-ness.	Gauge length.	0.1% Proof stress, tons per sq. in.	Ultimate tensile stress, tons per sq. in.	Elonga-tion, per cent.	Remarks.
A	.564″	—	1″	—	11.4	5.5	*
B	.5635″	—	1″	—	16.3	12.0	
C	.564″	—	1″	—	15.0	7.75	
D	.564″	—	1¼″	—	15.3	8.0	†
E	.5645″	—	1¼″	—	15.2	10.0	
F	.5635″	—	1¼″	—	16.4	12.0	†
G	.564″	—	1″	—	16.0	11.0	†
H	.564″	—	1¼″	5.0	15.2	8.0	†
J	.5645″	—	1¼″	5.0	15.0	8.0	†
K	.5635″	—	1″	—	16.5	12.25	†
L	.564″	—	1″	—	16.0	10.5	†
M	1.055″	1.341″	1″	—	13.9	13.0	
N	1.013″	.327″	1″	—	17.0	11.0	*
		Minimum		5.0	11.4	5.5	
		Maximum		5.0	17.0	13.0	
		Average		5.0	15.3	9.9	

* Microshrinkage at fracture. † Broke outside gauge length.

alloy (15.72) and the mean elongation is considerably greater. On the other hand it must be noted that the 0.1 per cent proof stress, 5 tons per square inch of the magnesium alloy is less than one half of the corresponding value (11.65 tons per square inch) for the aluminum alloy. When the Elektron casting illustrated was drawn at random from store, the job had been in continuous production for a sufficiently long period for the foundry technique to be considered as fully developed. It had been passed without comment as satisfactory to Class I standard after radiological examination. A larger casting in the same material, presenting greater difficulties of technique not only in the foundry but also in radiological examination, is the bracket illustrated in Figures 8a, 8b and 8c. In this case the overall dimensions were 28 inches × 19 inches × 8½ inches and the weight 21¼ pounds. The results of tests on samples cut from various parts of the casting are shown in Table VII. The effect of the

TABLE VII.

UNDERCARRIAGE BRACKET. MATERIAL: ELEKTRON A. 8—D.T.D. 289.

Bar.	Width.	Thickness.	Gauge length.	0.1% Proof stress, tons per sq. in.	Ultimate tensile stress, tons per sq. in.	Elongation, per cent.	Remarks.
A	.565″	—	2″	5.0	10.0	4.0	*
B	.754″	.422″	2″	—	12.9	4.0	* †
C	.865″	.237″	2″	—	14.4	7.0	†
D	.876″	.345″	2″	—	15.1	8.5	†
E	.565″	—	1½″	—	16.9	12.0	
F	.564″	—	1½″	—	14.0	8.0	
G	.564″	—	1″	5.1	14.8	8.0	
H	.565″	—	1″	—	12.6	7.0	
J	.5645″	—	2″	5.2	9.9	3.5	*
K	.565″	—	1½″	—	11.1	5.5	*
L	.565″	—	1½″	—	15.8	11.0	
M	.564″	—	2″	5.3	17.1	12.0	
N	.566″	—	1″	5.2	14.1	7.0	
O	.565″	—	2″	5.0	10.7	6.0	*
P	.895″	.230″	2″	—	10.6	4.0	†
Q	.876″	.400″	2″	—	10.0	3.0	* †
R	.500″	.360″	1½″	—	10.3	5.0	*
S	.5645″	—	1½″	4.8	12.6	6.5	
T	.406″	.396″	1½″	—	11.5	4.5	*
U	.565″	—	1½″	—	12.0	5.0	*

* Microshrinkage at fracture. † Broke outside gauge length.

Minimum	4.8	9.9	3.0
Maximum	5.3	17.1	12.0
Average	5.0	12.8	6.6

greater complexity of form is seen in the greater scatter and lower mean values of strength and ductility.

Some time has been devoted to these examples because the author wished to demonstrate the realistic basis on which we can now discuss the properties of the material in light alloy castings produced in circumstances where the technique of casting and of inspection are maintained at a reasonable standard. The performance of castings of this quality in aluminum and magnesium alloys as stressed components in aircraft has justified their serious consideration by the marine engineer.

Fatique Properties.

Examples have been described to illustrate the order of the variation in mechanical properties which may be expected in light alloy casting of relatively simple and also of complex form when a satisfactory foundry technique has been established. It is inevitable that the properties reported are generally determined in the ordinary tensile test, since more elaborate tests would call for more specimens than can usually be provided from a single casting, and a large number of castings is generally not available for sectioning. Opportunities occur occasionally, however, to obtain information on the fatigue properties of specimens cut from actual castings. W. Linicus and E. Scheuer* in 1934 reported the results of their own and other investigator's tests on various engine castings in aluminum alloys.

In Table VIII is a summary of Linicus and Scheuer's findings in respect of average tensile strength and fatigue strengths, generally at 20 million cycles, as determined on test pieces cut from the casting. For each material the corresponding values of tensile and fatigue strength in separately cast, standard test bars are shown. The castings investigated were crank cases, radial and in-line, of petrol and Diesel engines, ranging in weight from 16 to 350 kilograms, cylinder heads, and test samples of 25 millimeters diameter. In general the fatigue strength in the casting shows a smaller fall from the value in the separately cast test bar than occurs in the case of the tensile strength. This is brought out at the foot of Table VIII, from which a further condensed summary may be prepared as follows:

For all three alloys examined:—

$$\text{Ratio: } \frac{\text{Average tensile strength in casting}}{\text{Average tensile strength in test bar}} = 0.81$$

$$\text{Ratio: } \frac{\text{Average fatigue strength in casting}}{\text{Average fatigue strength in test bar}} = 0.91$$

It may be noted, however, that in the heaviest casting, the Diesel engine crankcase weighing 350 kilograms, section 35×40 millimeters, the fatigue strength was reduced to 0.73 of the test bar value, while the tensile strength ratio only fell to 0.80.

In the author's experience results in general keeping with the above have been obtained in large marine engine castings. In one example the opportunity occurred to make extensive mechanical tests on the material of a scavenge pump piston from a 4,000 Hp. double acting Diesel engine. This was a casting 60 inches in diameter in Ceralumin B alloy, the general form of which is shown in Figure 9. Four specimens were cut from the piston for tensile tests, two from the heavy section and two from the light. Seven test pieces, three from the thicker and four from the thinner portions, were taken for fatigue tests in Wöhler rotary bending machines, with single point loading. The results of the tensile tests are shown in Table IX and the stress-endurance fatigue curve in Figure 10. The requirements of the specification D.T.D. 287 covering aircraft uses of Ceralumin B. are:

Ultimate tensile stress ..	10 tons per square inch;
Elongated	2 per cent.

The values of tensile strength on a standard sand cast test bar reported on the four specimens taken from the casting, range from 82 to 103 per cent of the specification requirement, with an average of 94 per cent. A normal value for the fatigue strength of this alloy in a sand cast test bar, on a basis of 20-million reversals is ± 4.5 tons per square inch. The

* *Metallwirtschaft*, XIII, 47, 48, Nov., 1934.

TABLE VIII.
TENSILE STRENGTH AND FATIGUE STRENGTH IN CASTINGS

Material.	Casting.	Average ultimate tensile stress, kg./ sq. mm.	Fatigue 20 million cycles, kg./ sq. mm.
(1) Copper-silumin. Si. 12%, Cu. 0.8%, Mn. 0.3%, Fe. 0.5%	Aero Diesel crankcase 150 kg. 8–43 mm. sections	15.8 (10.04)	5.5 (3.49)
	Aero engine crankcase 42 kg.	14.0 (8.89)	5.5 (3.49)
	Diesel crankcase. 350 kg. Sections 35 × 40 mm.	15.5 (9.85)	4.0 (2.54)
	Separately cast test-piece, 12 mm. diam.	19.5 (12.36)	5.5 (3.49)
(2) Silumin gamma heat-treated. Si. 12%, Mg. 0.3%, Mn. 0.5%, Fe. 0.5%	Cylinder head 30 kg.	21.3 (13.52)	8.0 (5.08)
	Aero crankcase 42 kg.	21.3 (13.52)	7.6 (4.83)
	Aero crankcase 42 kg.	23.9 (15.16)	7.0 (4.45)
	Radial aero crankcase 16 kg.	27.2 (17.26)	7.5 (4.76)
	Separately cast test-piece, 12 mm. diam.	28.0 (17.77)	8.5 (5.40)
(3) Silumin beta	T piece, 25 mm.	15.1 (9.58)	6.0 (3.81)
	Separately cast test-piece, 12 mm. diam.	18.5 (11.75)	6.5 (4.13)
(4) Silumin gamma	T piece, 25 mm.		5.5 (3.49)
	Separately cast test-piece, 12 mm. diam.	20.5 (13.01)	6.5 (4.13)

(Values in brackets are tons per square inch.)

(1) $\frac{\text{Average U.T.S. in casting}}{\text{Test piece U.T.S.}} = \frac{15.1}{19.5} = 0.77$

$\frac{\text{Average fatigue in casting}}{\text{Test piece fatigue}} = \frac{5}{5.5} = 0.91$

(2) $\frac{\text{Average U.T.S. in casting}}{\text{Test piece U.T.S.}} = \frac{23.4}{28} = 0.84$

$\frac{\text{Average fatigue in casting}}{\text{Test piece fatigue}} = \frac{7.5}{8.5} = 0.88$

(3) $\frac{\text{Average U.T.S. in casting}}{\text{Test piece U.T.S.}} = \frac{15.1}{18.5} = 0.82$

$\frac{\text{Average fatigue in casting}}{\text{Test piece fatigue}} = \frac{6}{6.5} = 0.93$

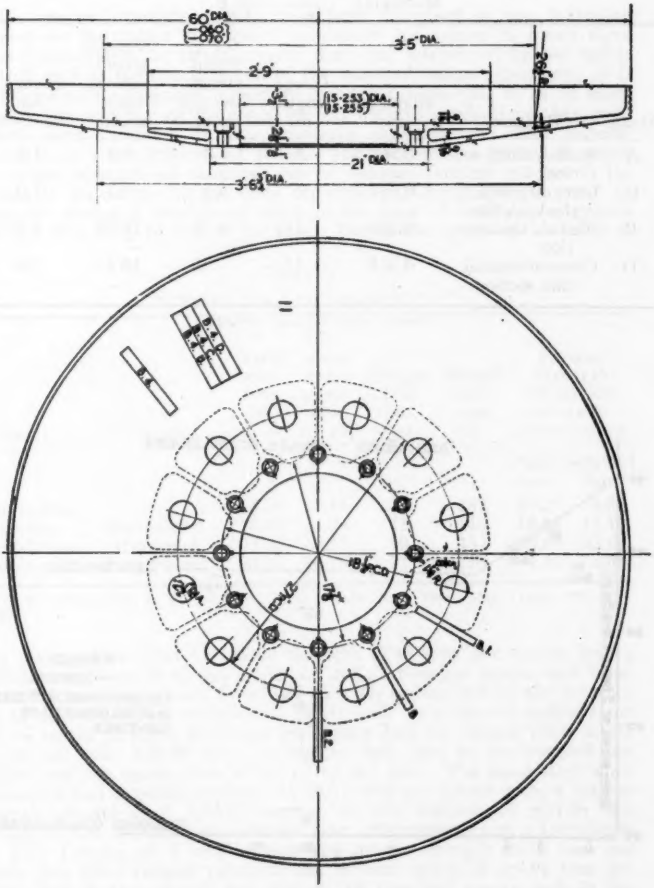
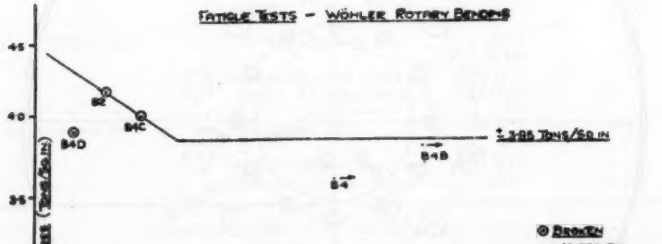


FIGURE 9.

TABLE IX.
MECHANICAL TESTS—SCAVENGE PUMP PISTON.
MATERIAL: CERALUMIN B.

Mark.	Position.	Diameter, in.	Gauge length, in.	0.1% Proof stress, tons per sq. in.	Ultimate tensile stress, tons/ sq. in.	Elongation, per cent.
A	Radial, thick section	0.564	2	6.8	9.0	1.5
C	Circumferential, thick section	0.564	2	6.6	8.2	1.0
B	Radial, thin section	0.358	1¼	6.79	10.08	1.3*
D	Circumferential, thin section	0.358	1¼	7.28	10.29	2.0

* Broke outside gauge length.



MATERIAL:- CERALUMIN 'B' FROM R.W. SCOVENDS PUMP PISTON CASTING
POSITIONS OF SPECIMENS SHOWN ON SEPARATE SKETCH

FIGURE 10.

figure of ±3.85 tons per square inch which the results clearly indicate for the fatigue strength of the material in the casting is thus 86 per cent of the value for the separate test bar.

Effect of Stress Concentration.

The fatigue strengths which have been discussed in the foregoing account are the values determined on smooth test pieces in which there is no concentration of stress greater than that introduced by the radius of 0.625 inch at the chucking-end of the standard fatigue specimen. It is fairly generally recognized nowadays that for fatigue data to be of much value in the selection of materials for engineering applications they must include some indication of their sensitivity to notches or other features causing local concentrations of stress. The very serious extent to which the fatigue strength of materials can be reduced through notches in the surface is shown by fatigue tests reported by Oberg and Johnson* on specimens having a standarized notch in the form of a 60 degree groove with a radius of 0.010 inch at the base. The results are given in Table X.

Table X.
Notched Bar Fatigue Results.*

Steel.	Treatment.	Yield point, tons per sq. in.	Ultimate tensile stress, tons/sq. in.	Elongation per cent.	Brinell hardness no.	Fatigue Strength 20 million reversals. ±tons/sq. in.	
						Plain bar.	Notched bar.
Plain carbon	—	56.70	57.45	20	257	35.27	13.84
Chrome-Molybdenum	Normalized	20.09	34.01	32	145	18.75	11.16
	Hardened	75.23	88.97	10	432	50.89	23.66
Duralumin	Heat treated	20.98	26.78	12	120	4.91	2.68

* T. T. Oberg and J. B. Johnson, Proc. Amer. Soc. Test. Mat., 1937, 37, (II), 195-203.

In a plain carbon steel of tensile strength 57.45 tons per square inch a fatigue strength at 20 million cycles of ±35.27 tons per square inch in an unnotched bar was reduced to ±13.84 tons per square inch by the presence of a 60 degree notch, a reduction of 54 per cent. In a chrome molybdenum steel of tensile strength 88.97 tons per square inch the fatigue value in the smooth bar was ±50.89 tons per square inch, and in the notched bar ±23.66 tons per square inch, a fall of 61 per cent. The same steel when normalized had a tensile strength of 34.01 tons per square inch, a fatigue strength unnotched of ±18.75 tons/sq. in. and notched of ±11.16 tons per square inch, a reduction of 42 per cent. Specimens from a Duralumin (27 ST) forging of 6 inch section and tensile strength 26.78 tons per square inch gave fatigue values at 200 million cycles of ±4.91 tons per square inch in the smooth bar, and ±2.68 tons per square inch in the notched bar. It will be noted that the reduction in fatigue strength caused by the presence of the notch, 57 per cent, is of the same order as that in the stronger steels.

In cast aluminum alloys the published data available indicate that the notch sensitivity is in general less than in the wrought alloys. Results due to Linicus and Scheuer* gave a reduction of 20 per cent in the

* *Metallwirtschaft, XIII,* 47, 48. Nov., 1934.

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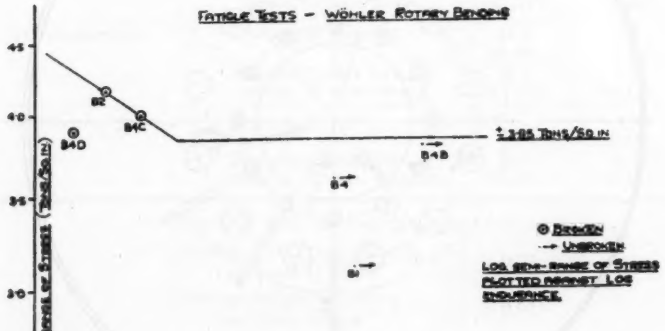

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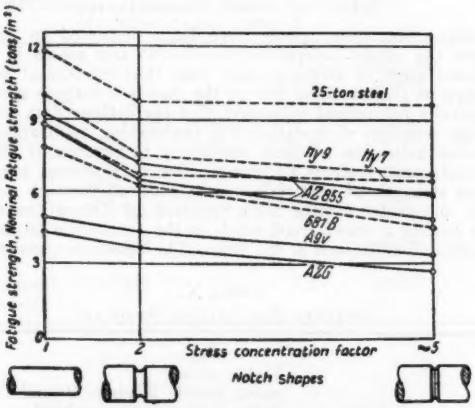
* *Metallwirtschaft, XIII,* 47, 48. Nov., 1934.

endurance value of silumin and silumin gamma, while Armbruster† reported reductions varying from 7 to 22 per cent, caused by the irregularities of the cast skin.



Effect of notch shapes with different stress concentration factors (bending stress) on the nominal fatigue strength of Elektron AZ855 and sand-cast Elektron AZG and A9v. 25 ton steel, Hy 7 (Al + 7% Mg), Hy 9 (Al + 9% Mg) and duralumin 681B (Al + 4% Cu, ½% Mg) are shown for purposes of comparison. (Tests by W. Buchmann, 1936.)

FIGURE 11.

Figure 11‡ shows in a summarized form the effect of two different types of stress raising notches on the fatigue strength of the magnesium alloys, wrought and cast, in most general use, and of three aluminium alloys and a 25 ton steel. The view is held in some quarters that magnesium alloys are exceptionally sensitive to the effect of notches under dynamic loading, but the examples shown suggest that there is not a great deal of difference in sensitivity between the magnesium alloys and at least the stronger of the aluminum alloys.

The significance of these observations is that forgings and castings in aluminum alloys are not more sensitive to the effect of notches than are the ferrous materials with which the marine engineer is familiar. At the same time the necessity remains for the designer and manufacturer to see that there are no avoidable notches, sharp-angled shoulders or abrupt changes of section, whatever the material of construction.

INTERNAL STRESSES IN LIGHT ALLOY CASTINGS.

Internal stresses in light alloy castings intended for engineering applications may be objectionable for two main reasons. The first is the distortion which is liable to occur when a casting having internal stresses is machined, with consequent release of stresses. Deformation of this nature is troublesome when it is desired to machine the castings to close

† Quoted by Linicus & Scheuer from Diss. T. H. Munich, 1930.

‡ Technology of Magnesium and its Alloys, p. 216, Adolf Beck. Translated and published by F. A. Hughes & Co., Ltd., Abbey House, London, N.W.1.

limits of accuracy. The second reason is that the internal stresses may represent a dangerous supplement to the stresses imposed by the external loads of normal service.

Recognized sources of internal stresses in castings are non-uniform cooling of the casting in the mould following solidification, non-uniform cooling in heat treatment and changes in volume accompanying constitutional transformations.

The determination of the magnitude of the internal stresses in castings, especially if they are of irregular form, is a matter of considerable difficulty and uncertainty. It is possible, however, to obtain guidance as to the effect of various thermal treatments on the internal stresses by a fairly simple experimental process. This may be illustrated by reference to results obtained with cast Y-alloy.

The interest in stress-relieving cast Y-alloy pistons has already been mentioned, and the results to be described were obtained during an investigation made some years ago to establish the heat treatment which would leave the internal stresses at a minimum while producing the best hardness. To provide material for this purpose, cylinders of Y-alloy were cast in sand moulds to the dimensions: outside diameter 9⅛ inches, inside diameter 7⅝ inches, length 10 inches. From these cylinders rings of o.d. 8¾ inches, i.d. 7¾ inches, length 1½ inches were machined. After subjecting each ring to the desired treatment, gauge marks were lightly punched 2 inches apart on a flat face and the distance between them accurately measured by means of a traveling microscope. The ring was then clamped to the table of a milling machine and the ring was cut through at a point midway between the gauge marks, using a cutter ⅛ inch wide. The clamp was released and the distance between the gauge marks again measured. The amount of movement thus revealed was taken as a measure of the relative magnitude of the internal stress. The results observed, together with the Brinell hardness (10/1000/30) are given in Table XI. The most important inference from these data is that if boiling water is used as the quenching medium the internal stresses are consider-

TABLE XI.

INTERNAL STRESS IN SAND CAST Y-ALLOY CYLINDERS.

Mark.	Condition.	Change, inch.	Brinell Hardness, 10/1,000/30
Y S 1	As cast	−.0021	84.9
Y S 2	Annealed 2 hours at 450° C. and cooled with furnace.	−<.0002	46.0
Y S 3	Heat treated 6 hours at 530° C., quenched in boiling water, aged 11 days at room temperature.	+.0020	88.4
Y S 4	Heat treated 6 hours at 530° C., quenched in cold oil, aged 11 days at room temperature.	−.0185	93.7
Y S 5	Heat treated 6 hours at 530° C., quenched in boiling water, tempered 16 hr. at 175° C.	−.0007	114
Y S 6	Heat treated 6 hours at 530° C., quenched in cold oil, tempered 16 hr. at 175° C.	−.0215	114

ably lower than when the castings are quenched in oil. When quenching in boiling water is followed by tempering for 16 hours at 175 degrees C., the internal stresses appear to be brought down to a very low value, while the hardness is as high as when the casting is quenched in oil before tempering. The internal stresses are evidently at their lowest after annealing at 450 degrees C. and slowly cooling from that temperature, a treatment which produces the softest condition.

Guidance in the selection of the temperature and duration for a stress-relieving treatment appropriate to cast light alloys may be obtained by employing specimens in which an internal stress has been deliberately induced. In one method a ring of the material under consideration is cast and machined in a similar manner to the Y-alloy rings previously described. In the further treatment of the rings, however, a different course is followed. A gap of carefully measured width is cut through the ring. By introducing wedges of successively increasing thickness into the gap the material of the ring is strained to correspondingly increasing degrees. Figure 12 shows a ring before and after insertion of the wedge. On withdrawing the wedges and measuring the gap by means of feeler gauges the maximum elastic strain is found. The procedure then is to insert into the gap a wedge producing a strain short of that causing a permanent set and to submit the ring, with the wedge in position, to the desired thermal treatment. When the ring is cool the wedge is removed and the width of the gap remeasured. If the ring springs back so that the width of the gap is no different from what it was originally, it can be taken that the treatment applied would not be effective in relieving internal stress. On the other hand, if the edges of the gap do not close in when the wedge is removed it is an indication that the treatment would be completely effective in removing internal stress. Intermediate degrees of movement on extracting the wedge imply that the internal stresses have been partially removed.

By experiments on these lines it is found that in a aluminum-silicon alloy castings treatment for 3 hours at 300 degrees C. followed by normal free cooling in air removes the greatest internal stresses which could be present in a casting. This treatment causes a slight reduction in the proof stress and tensile strength—less than 0.5 tons per square inch in both properties—and a corresponding increase in ductility. The degree of softening which the values represent would not affect the serviceability of the castings in any readily imaginable circumstances.

In alloys which depend for their desirable mechanical properties upon heat treatment, the effect of a stress-relieving treatment on the strength and hardness has to be considered rather closely. In castings of Ceralumin B, for example, which are normally heat-treated for 8 hours at 165 degrees C., followed by air cooling, the proof stress in tension is appreciably reduced by any subsequent heat-treatment which removes internal stresses substantially or completely. If the change in the width of the gap on extracting the wedge after heating and cooling is assumed to be directly proportional to the remaining internal stress it can be said that while treatment for 2 hours at 330 degrees C. removes 95.2 per cent of the induced stress, heat treatment for 5 hours at 250 degrees C. only removes 67.2 per cent. On the other hand, the treatment at 250 degrees C. leaves the 0.1 per cent proof stress at 7.35 tons per square inch, while after heating for 2 hours at 330 degrees C. the proof stress is reduced to 5.16 tons per square inch.

In magnesium alloy castings internal stresses are generally smaller than in aluminum alloys; this is to be expected in view of the lower creep resistance and the lower tensile strength of magnesium alloys at elevated temperatures. The times and temperatures appropriate for stress-relieving of unheat-treated aluminum alloy castings are effective also for magnesium alloys. The heat-treatment of magnesium alloy castings does not neces-

sitate quenching, and in consequence "solution-treated" Elektron A 8 castings, i.e., castings which have been heated for say 16 hours at 400 to 420 degrees C., are practically free from internal stress.

Conclusion.

In this paper the author has tried to present some of the characteristics of light alloys which it seems designers will wish to take into account when deciding what rôle these materials can play in the further development of the marine engine. It is typical of the principle to which the metallurgist strives more and more to refer his selection and treatment of metals and alloys, that attention has been directed to properties which are not included in the usual specifications governing the inspection of castings and wrought products, but which are of even greater significance in relation to their probable behavior in service. To know the nominal properties derived from a standard test piece produced under prescribed conditions is not so interesting as to have some idea of how these properties are affected by variations of section, surface imperfections, changes of temperature and the other factors which make up service conditions. It is the responsibility of the metallurgist to provide this information; in many cases it can be given from existing knowledge, in others experimental investigation is necessary. So far as light alloys are concerned, large programs of testing are either in progress or are being planned in various establishments. It is important that marine engine designers who are interested in the possible application of light alloys to their own problems should make known on what features of the mechanical and physical properties of these materials they consider further information is required. In the present paper the author has made a guess at some of the headings under which that information might be considered, but his treatment of the subject has necessarily been incomplete. He will feel that his remarks have served a useful purpose, however, if they stimulate a discussion which will help the metallurgist to appreciate the attitude of the marine engine designer towards light alloys and his needs as regards technical information.

NEW WARSHIP CONSTRUCTION.

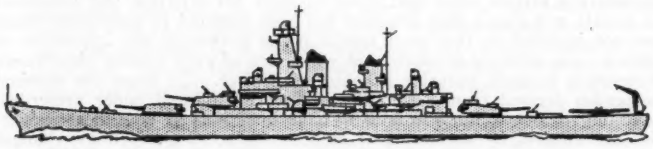
This article by "Navarino" is reprinted from the November, 1944 issue of the Journal of the Royal United Service Institution.

With the publication of the 1943-44 edition of *Jane's Fighting Ship* the time is ripe, and the means to hand, for our annual review of new warship construction; but the expansion of the two principal Allied Navies, even in the past year, has been so immense that it will be necessary to devote most of our space to them, and to make brief references only to any important new additions to enemy fleets.

Battleships.

Five years of war against our Eurcpean enemies have been remarkable for the fact that they have witnessed no encounter between the main fleets. Nevertheless the battleship has proved her versatility and utility to a degree which must confound her critics. She has engaged in duels with her own class in mid-Atlantic and in the darkness of the Arctic Ocean; she has taken part in a night action against cruisers in the Mediterranean; and she has been used boldly for bombardments to cover major landings and to assist in the capture of heavily defended ports. Her heavy armament, stout protection, and—in the latest types—high speed, have all proved of value.

It is interesting to note that in the course of a Discussion on "Speed and Gun Power in Warships" held in the Institution in March, 1938, a design for a battleship to carry nine or ten 14 inch guns and to have a speed of 32 knots was condemned as being impracticable. Only two years later H.M.S. *King George V* was completed with the higher number of those guns and a speed of 30 knots; and now we see the U.S. *Iowa* class with nine 16 inch guns and a designed speed of 33 knots, which is unofficially reported to have been exceeded.



U.S.S. "Iowa".

The two first ships of the *Iowa* class are completed, and are remarkable as being the biggest warships that have ever been built: they displace 45,000 tons (52,000 full load). In addition to their main armament, they carry twenty 5 inch dual purpose guns and over 100 light A.A. and machine guns. Four more of the same class are shown as building.

No details are given of their British contemporaries—the four ships of the *Lion* class; but they are believed to be about the same displacement and to have a similar armament.

Orders were placed in 1940 for five battleships for the American navy of a still larger type, to displace 58,000 tons (65,000 tons full load); but it is doubtful whether they have got beyond the blue print stage.

The only other battleships on the allied side worthy of comment are the French *Richelieu,* which has been thoroughly refitted and partially re-armed in an American yard, and her sister-ship—the *Jean Bart,* which, after being badly knocked about at Casablanca, is understood to be undergoing a similar rejuvenating process.

Germany having squandered her battleships by using them singly or in pairs as surface raiders, is ending the war without a single capital ship fit for action. Two sister ships of the sunken *Tirpitz* are shown as building, but *Jane* remarks that it is doubtful whether they will ever be completed. For the rest, there remain afloat the badly damaged *Gneisenau* and the original so-called pocket (10,000 ton) battleships *Lützow* and *Admiral Scheer,* the first of which was torpedoed, but not sunk, by our midget submarines in September of last year.

Very little reliable information is available about Japan's five new battleships, but it seems probable that they are more or less contemporaries of the U.S. *Iowa.* It is believed that they have all been laid down and the first three may have been completed. The sister ships *Huso* and *Yamisiro,* reconstructed in 1932-33, were both sunk in the battle of the Philippines in October of this year.

Battle Cruisers.

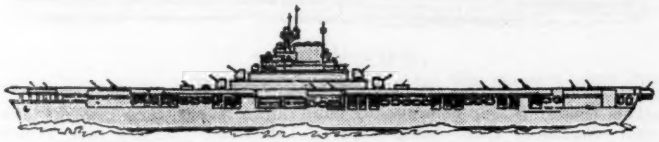
The battle-cruiser type is not quite dead. Unofficial details of the six U.S. ships of the *Alaska* class describe them as displacing about 27,000 tons and mounting six 14 inch and sixteen 5 inch guns; their designed speed may be as high as 35 knots. Two have been launched, but it seems doubtful whether the others will be completed.

These ships seem to have been intended to be a counter to the four new Japanese armored ships. The latter appear to have been modelled on the German "pocket battleships," and are believed to displace 12,000-15,000 tons and to carry six 12 inch guns, perhaps in triple turrets. They may, however, be completed as aircraft carriers.

Aircraft Carriers.

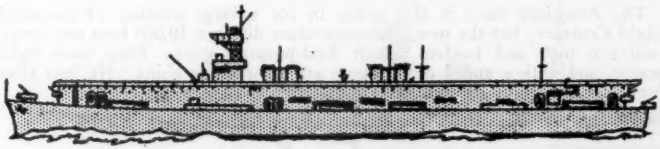
Another much criticized type which has more than made good in this war is the aircraft carrier. In her various shapes and sizes she has taken part in many different forms of operations. She looks like being the ship of the future for long distance attacks against an enemy's main fleet; she has already proved indispensable as a floating and highly mobile base for aircraft supporting a landing or attacking harbor and other objectives out of range of airfields; as an escort for convoys and a counter to submarine attacks she is invaluable.

Nothing newer than the *Indomitable* class of the Fleet Aircraft Carrier type is shown in the British section of *Jane;* but it is evident that the number of our escort carriers of the 14,000 ton and of a smaller type is increasing steadily: some forty ships of these classes are mentioned by name.



U.S.S. "Essex."

The U.S. Navy has gone in for carriers in a really big way, and about a hundred are in commission and many more under construction. At the head of the list are three ships of 45,000 tons designed to carry very large bombers—two ordered in October, 1943, and one due to be laid down early this year. Next come seventeen named ships of the *Essex* class of 27,000 tons and 35 knots, the majority of which are already complete. It is noteworthy that the first was built in 20 months, the second in 17½ months. Each carries some hundred aircraft. A smaller type are the nine *Independence* class of 10,000 tons carrying fifty or sixty aircraft.



U.S.S. "Independence."

Four of the older carriers have been sunk in the war with Japan, and only the *Enterprise, Ranger* and *Saratoga* remain of the pre-war veterans.

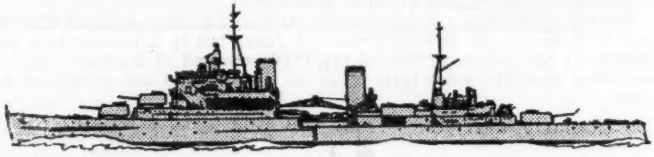
When we come to the invaluable Escort Carriers, we find that their names and numbers are legion.

Japan has lost at least ten carriers, including four sunk in the battle of

the Philippines, and so far as information goes does not seem to have made any great efforts to replace them. Doubtless reliance was originally placed on the island air bases which she seized early in the war, but from which she is being steadily ejected.

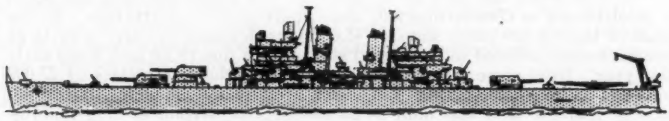
Cruisers.

The latest British cruisers about which details have been released are the three ships of the 8000 ton *Uganda* class. They are of similar displacement to the earlier *Mauritius* class, but have only one after triple turret, and mount nine, instead of twelve, 6 inch guns. Alterations to the original 10,000 ton 8 inch class have been many and extensive, but none have been so drastic as those in the *London*, which, with her two instead of four funnels, now looks like a large *Mauritius*.



H.M.S. "Uganda."

The United States continue to build large 8 inch gun cruisers, the latest—*Baltimore* class—being ships of 13,000 tons carrying nine 8 inch and twelve 5 inch guns. This is no heavier armament than that of the earlier class of under 10,000 tons; but better armor appears to account for the additional displacement.

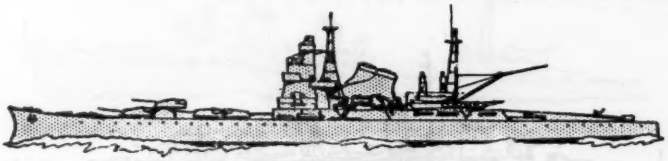


U.S.S. "Baltimore."

The American navy is also going in for a large number of so-called Light Cruisers; but the new *Cleveland* class displace 10,000 tons and carry twelve 6 inch and twelve 5 inch dual-purpose guns They have light armor, and with a speed of 33 knots are formidable ships. No less than forty-one are built or building.

U.S.S. "Cleveland."

The Japanese opposite numbers to the American "light" cruisers are their "Second Class" cruisers. The most modern of these are the two ships of the *Tone* class. They displace about 10,000 tons and are believed to carry eight or more 8 inch, or possibly twelve 6 inch, guns, and to have a speed of 33 knots.



"TONE" CLASS (JAPAN)

The latest Japanese light cruisers are referred to in a Special Late Addenda as the *Agano* class of 6000 tons and carrying six 6.1 inch guns. They have one funnel and a heavy bridge. Casualties have seriously reduced the cruiser strength of the Japanese navy, and six more of the heavy and three of the light type were sunk in the battle of the Philippines.

MONITORS.

The value of naval bombardments with the heaviest guns has been emphasized again and again in this war, and it has been found that in certain circumstances bombing from the air is not an effective substitute for this form of attack.

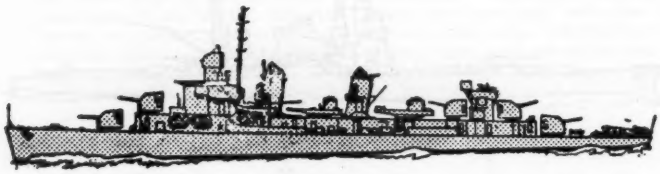
In the absence of a considerable number of monitors, such as we possessed in the last war, battleships have had to be employed on this service. Fortunately, we found ourselves with a plethora of them when the time came; but after the War there is bound to be a demand to reduce the number of capital ships in favor of other types, particularly carriers. Then the Coast Attack ship may come into her own again as a standard type of warship which must be maintained in a peacetime navy. Meanwhile the sole remaining monitor from the last war—the *Erebus*—has been much in the limelight in her work on the French coast. The two new ships of this class—the *Roberts* and *Abercrombie*, which appear in *Jane* for the first time, have also been mentioned in despatches.

H.M.S. "ROBERTS."

DESTROYERS.

With the passing of great battle squadrons and their array of satellites, the destroyer is becoming less and less a main fleet unit and more and

more a "maid of all work." Lacking sufficient Coast Attack ships, she has even had to be pressed into service for bombarding. Doubtless her speed is an asset for "tip and run" attacks or for getting her out of a tight corner; but the 4.7 inch gun can scarcely make up in numbers what it lacks in weight of projectile—as compared with the monitors' 15 inch (or even the 9.2 inch of last war) for this work.



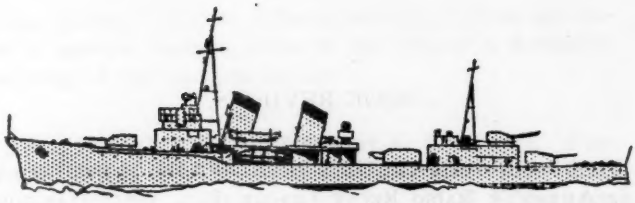
U.S.S. "FLETCHER."

If there are already fewer battleships needing destroyer escorts, there are far more aircraft carriers to claim their protection, quite apart from other extensive calls for their services. The U.S. Navy particularly has kept pace with this new requirement, and now has an enormous number of destroyers. The latest are of three designs—one with a nominal displacement of 1700 tons (about 2000 full load) and carrying four 5 inch guns and five 21 inch tubes; the second—the numerous *Fletcher* class—of 2100 tons (about 2500 full load), with five 5 inch guns and ten 21 inch tubes; and the third of 2200 tons with six 5 inch guns. The designed speed of the smallest class is 36.5 knots; that of the two larger is believed to be a good deal higher.

H.M.S. "QUEENBOROUGH."

Our much more modest program of destroyer construction seems to have concentrated on a "utility" type, the standard armament of which is, so far, four 4.7 inch and apparently four 21 inch tubes. No details of speed are given.

Japan has, for some years, been adding rapidly to her destroyer flotillas, and *Jane* believes that new ones are being built at the rate of about twenty a year. The latest design, of which particulars are given in the Special Late Addenda, displaces over 2,000 tons and carries the heavy armament of eight 5 inch guns and four 21 inch tubes. The prolific earlier class of slightly smaller ships carry six 5 inch and eight tubes. Their speed is reported to be 36 knots.



"KAGERO" CLASS (JAPAN).

No new destroyer since those of the 1938 program are mentioned in the German section.

ESCORT SHIPS.

Escort ships—Sloops, Frigates and Corvettes—have sprung up like mushrooms in this war, and have played a great part in winning the Battle of the Atlantic.

The latest British sloops of which particulars are given displace 1250 tons; carry six 4 inch A.A. guns; and have a speed of about 19 knots.

The *Captains* class frigates displace 1300 tons; carry three 3 inch dual purposes guns; and have a speed of about 20 knots. They are being turned out in large numbers from American yards by the pre-fabrication system.

Corvettes seem to owe their origin to the Tees; but in Canada they are turned out cheaply and quickly in large numbers. On a displacement of 925 tons, they carry one 4 inch and a number of smaller guns, and have a speed of 17 knots.

The U.S. Navy now possesses over seven hundred "destroyer escorts" —similar to our *Captains* class frigates and nearly eighty "frigates rated as escort vessels."

MISCELLANEOUS WARSHIPS.

Space will not permit of even a superficial review of the many other types of warships which will be found enumerated in *Jane*—Submarines; Seaplane Tenders; Minesweepers; Minelayers; Motor Torpedo Boats; Motor Gunboats; Armed Trawlers; Landing Craft; and Fleet Auxiliaries: they are all playing their part in the complex business of the war at sea.

BOOK REVIEWS.

THE RADIO AMATEUR'S HANDBOOK. Published by the American Radio Relay League, Inc., West Hartford, Connecticut. $1.00.

The 1945 issue of this well known handbook, the twenty-second edition, is now available. Since 1926, the book has grown steadily through the years—in size, in value, in acceptance. This edition is a comprehensive, authentic digest of the older, familiar phases of radio theory and practice, interspersed with forecasts of the more advanced techniques of tomorrow.

A NAVAL LOG, By Commander W. K. Thompson, U.S.N.R. and Lieutenant G. W. Stone, Jr., U.S.N.R. Published by D. Van Nostrand Company, Inc., 250 Fourth Avenue, New York, N. Y. $2.50.

This is a compact handbook of essential information for naval officers, compiled primarily from existing Navy Department publications. It contains pertinent facts concerning pay accounts, legal assistance, medical assistance, the many rights and privileges to which an officer and his family are entitled, the opportunities for further training in the Navy, movement of household effects, wardroom etiquette, and a host of other subjects.

DIESEL FUEL INJECTION SYSTEMS, By Louis R. Ford. Published by Diesel Publications, Inc., 192 Lexington Ave., New York 16, N. Y. $1.00 in the U.S.A., $1.25 Elsewhere.

Several articles describing the fuel injection systems of various makes of engines, previously published in the pages of the magazines "Diesel Power" and "Motorship", have been assembled in this small pocket-size volume. The material in the book is based upon information supplied by the manufacturers of the various

injection systems. Thirteen different injection systems are described in separate chapters, while the last chapter is devoted to the servicing of fuel injection nozzles.

DIESEL LUBRICATION, EDITED BY A. B. NEWELL. PUBLISHED BY DIESEL PUBLICATIONS, INC., 192 LEXINGTON AVE., NEW YORK, N. Y. $1.00 IN THE U.S.A., $1.25 ELSEWHERE.

The information contained in this 126 page, paper-bound book comes largely from the pages of "Motorship" in which appeared a series of articles on lubrication.

STEAM AND GAS TURBINES, BY STODOLA AND LOWENSTEIN. PUBLISHED BY PETER SMITH OF NEW YORK, IN TWO VOLUMES.

This standard text and reference of 1565 illustrations and of folding charts is based upon a translation of the sixth German Edition.

The volumes assume a greater significance than ever with the current and future status of the development and application of the gas turbine.

NOTES AND CORRECTIONS.

PAPER: "CALCULATION OF THE IMPEDANCE PROPERTIES OF PARASITIC ANTENNA ARRAYS INVOLVING ELEMENTS OF FINITE RADIUS." PUBLISHED IN THE MAY, 1945 ISSUE OF THE JOURNAL.

Equation 4 should read

$\Pi = 4\pi \times 10^{-7}$ henry per meter.

Π, the magnetic constant of space, should be used in lieu of π in (3), and in (23) to (26). Equation (33) is

$$R_c = \sqrt{\frac{\Pi}{\Delta}} = 376.7 \backsimeq 120\pi \text{ ohms.}$$

A negative sign should precede S_1' in (12). In (38) a_{12} should be replaced by a_1^2.

In analyzing an array numerically, it is probably better to retain all terms in (56) and (57) when computing C_1 and C_2, rather than use the simplified formulas (63) and (64). If this is not done, bothersome infinities may be encountered when radiators of special lengths are employed. Details of such calculations are given in a paper entitled "A Theory for Three-Element Broadside Arrays", now being considered for publication in the Proceedings of the IRE.

The integrals (46) to (49) are tabulated in Reference 2, for two parallel antennas of different dimensions.

A graphical representation of the theory has been carried out by the writer for similar antennas, and appears in the new McGraw Hill book "Transmission Lines, Antennas, and Wave Guides."

ASSOCIATION NOTES.

Secretary-Treasurer.

Captain F. W. Walton, U. S. Navy, has been appointed by the Council as Secretary-Treasurer of the Society for the unexpired term of Commander Robert T. Sutherland, Jr., U. S. Navy, resigned because of detachment from duty in Washington, D. C.

Commander F. B. Schultz has been appointed Assistant to the Secretary-Treasurer, vice Captain Walton.

Addresses.

Correct addresses are becoming more and more important every day. Please keep us advised so that you may be sure to get your Journal.

Membership.

The following have joined the Society since the publication of the May, 1945, Journal:

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